POWER PLANT ENGINEERING

Established Over 29 Years

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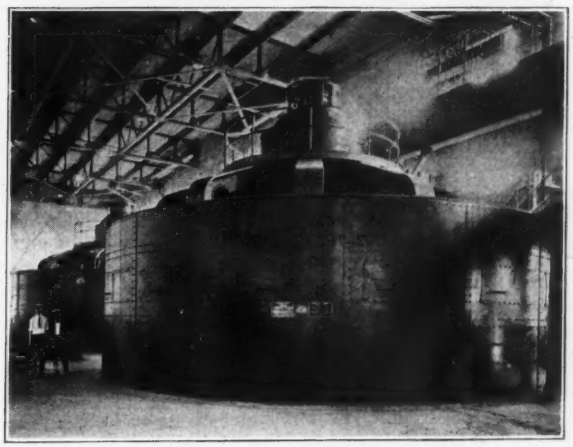


A Twentieth Century Giant

IN THE FAIRY tales of our childhood years we were fascinated by vivid accounts of giants: giants that stalked over mountain and plain with seven league strides and snapped big oaks across their knees as though they were twigs. At one time, before the coldness of sound logic had chilled the golden fancies of our childish imaginations, these giants were very real; we believed in them. But these fancies did not endure. As we grew older we found, not without a feeling of regret perhaps, that the heroes of the fairy tales were myths. There were no giants and we relinquished our belief in them.

Today, however, there are real giants; far more powerful than those of our childhood fancies. The one pictured above, for instance, is an electrical giant who has the strength of 96,000 horses. It is the second of two 65,000 kw. generators installed last year at the Niagara Falls Power Co. and its completion marks one phase of progress in the electrical industry. Other accomplishments in the electrical art during 1924 are described in an article on page 195.

IN A CERTAIN boiler plant, the rapid accumulation of carbon and slag on the rear arch of one of the furnaces was viewed with some concern. The boiler could not be shut down, however, so the deposit of slag was permitted to continue. To the surprise of the operators, instead of being detrimental to operation, the slag deposit improved matters, so much in fact that as explained on page 177, it pointed the way to better furnace design.

8000 B. Hp. for Standard Oil Co.

Two Types of Traveling Grate Stokers Installed Which Are Burning Small Size Anthracite

TO CHANGE OVER from the operation of 55 miscellaneous boilers to the operation of eight modern boilers has been the novel experience at the Bayonne plant of the Standard Oil Co. As explained by E. K. Knowlton, power superintendent, the old boilers were grouped in several boiler houses at various points throughout the plant. They were all oil fired and of various types including both water-tube and fire-tube. The old boilers ranged in size from 100 to 250 hp. and were normally operated at 120 per cent of rating. The new boiler house contains only eight water-tube boilers, which are often operated at 230 per cent of rating.

In addition to this new plant, there are still operated four other plants, which are located advantageously about the property, one of these being equipped for producing electric current by means of gas engines and steam turbines. It is from this source that electric power is derived for driving some of the auxiliaries in the new boiler plant. Current is generated and transmitted at 2300 v., 3 phase and 25 cycles. In the new boiler plant are house transformers, which step the voltage down to 440 and 110, depending upon the service for which it is to be used. The electric light circuits are operated on 110 v.

As a visitor approaches this plant, he will be impressed, first, by the manner in which coal is handled in the yard. Coal is brought to the yard in railroad cars from which it may be dumped directly into either of two track hoppers, which are located beneath adjacent tracks. Two feeder conveyors deliver the coal from the track hoppers to a 30-in. belt conveyor, the capacity of which is 275 T. per hr. This belt conveyor, which is in two sections, is constructed on a long incline so that it is used as a means of delivering the coal to a point over the coal bunker, which is located above the boiler firing aisle. The fact that all coal burned is anthracite, generally No. 3 buckwheat, explains why no crushers have been necessary.

Coal storage problems have been solved by the use of a drag scraper which is operated by means of an electrically driven winch located near the track hoppers. The counterweighted car, which acts as an outer snatch block for the cable, is operated on a circular track so that the coal is stored over an area of a semi-circle, the radius of which is approximately 280 ft. With coal piled 14 ft. high, the storage capacity within this semi-circle is 40,000 T. The drag bucket is operated between the car and an anchor block, which is located near the track hoppers. One interesting feature of this coal handling and storing arrangement is that there is but one pulling drum, which is used both for storing coal and reclaiming. The bucket is attached to one line of the cable if coal is to be stored and to the other or return line if it is to be reclaimed.

Coal to Be Stored Is Bypassed from the Conveyor

All coal, whether it is to be stored or not, is unloaded into the track hoppers and the feeders then carry it to the first or lower run of the belt conveyor, which elevates it to the beginning of the second run of the belt conveyor. At this junction point of the two belt conveyors is a gate and chute arrangement, whereby all or a part of the coal may be by-passed and dumped within the area covered by the drag scraper. It is the coal from this bypass chute which is spread out over the storage area. In reclaiming the coal, the drag scraper unloads into a chute, which delivers the coal to the feeders and it then goes to the belt conveyors, which deliver it to the boiler house.

Before entering the boiler house, it would be well to note here the two steam lines which leave at opposite ends of the boiler house and which serve the requirements of a part of the refinery. One of these lines is 4070 ft. long. It is 16 in. in diameter and carries high-pressure, superheated steam. The other line is also 16 in. in diameter, 3200 ft. long, and carries high-pressure steam. All of this construction is of the overhead type, the pipe being well insulated and the insulation protected from the elements by a sheet iron cover, which is held in place by bands. The covering laps are not cemented but the iron is kept painted with a good structural paint.

Due credit for the construction work on these lines must be given to the pipe fitting department of Bayonne Works, for when steam was turned into them for the first

time only two leaks were found in the hundreds of Vanstone joints of the long line.

At the point where the outdoor belt conveyor enters the boiler house is a distributing conveyor, which runs the full length of the 1500-T. coal bunker, which is located directly over the center firing aisle. This distributing conveyor is equipped with a tripper or unloader so that coal may be discharged from it at any point within its length. The coal bunker, just referred to, is of the suspended type and is made of steel and is lined Bitumastic. Coal gates are spaced on 9 ft. centers so that there is practically no dead storage in the bunkers and coal may be drawn off to load the weigh larries with a minimum of travel by the larries. These larries, of which there are two, are 5 T. capacity each and are arranged with two spouts so that coal may be discharged to either side of the firing aisle.

All the coal handling equipment was furnished by the Guarantee Construction Co. and it is all driven by General Electric Co. alternating-current motors.

Two Types of Stokers Were Installed

Mounted on each stoker is an extended hopper of about 15 T. capacity. The present arrangement of boilers is four on each side of the firing aisle. The four boilers on one side are fired by Illinois chain grate stokers and on the opposite side by Harrington traveling grate stokers, the latter being furnished by the Riley Stoker Corp. These stokers are all of the forced blast type and are approximately 18 ft. long by 16 ft. wide. Each stoker is driven by an individual Troy engine, the size of which is 5 by 7 in. The speed of the engines is controlled by hand; no automatic control devices have been installed as yet.

Engines driving the Harrington stokers are direct connected at the front end. The engines which drive the Illinois stokers are belt connected to individual shafts,

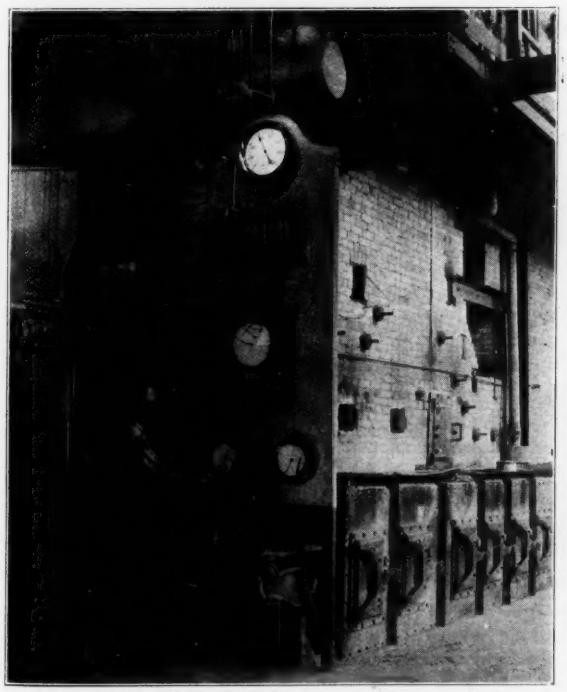


FIG. 1. EACH BOILER SETTING IS EQUIPPED WITH AN INDIVIDUAL GAGE AND INSTRUMENT BOARD

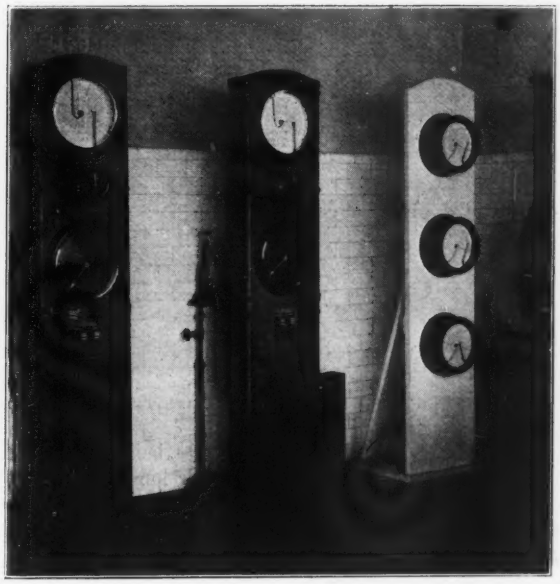


FIG. 2. FIVE RECORDING INSTRUMENTS ARE MOUNTED IN THE CHIEF ENGINEER'S OFFICE

which are equipped with two sprocket wheels so that each stoker is driven through two chains, one at each side of the front end of the stoker. Of considerable interest in this installation are the blower units which consist of two Coppus blowers for each boiler. They are turbine driven and one is located on each side of the setting and connected to a blastbox, which extends along the outside wall of the setting. These blastboxes are of steel and are independent of the setting except where air connections are made to the stoker. Air is drawn in over the top of the breeching where it is heated to about 100 deg. F. and thus some heat is reclaimed and discharged forward under pressures which may vary up to 5 in. of water. Each blastbox is equipped with suitable doors for inspection purposes. Damper controls to the stoker compartments are brought out to the side on the Illinois stokers and on the Harrington stokers they are brought out to the front of the setting.

Boiler settings are built entirely of fire brick except for some carborundum brick, which is used at the clinker line on the sides of the stokers and on the sloping shelf where the hot ash strikes the setting as it falls off the rear end of the stokers. The arches used are of an interesting design in that the front and rear arches come within about 4 ft. of each other near the center or throat of the furnace. The front arch extends back from the inside of the front wall about 8 ft. and 8 in. The rear arch extends forward so that a passage is left for the gases, which is 4 ft. wide and which extends across the width of the furnace. After passing this throat the gases expand again to enter the lower part of the first pass, which is 7 ft. wide. American suspended arches were used throughout.

On four of the furnaces some trouble was experienced due to warpage of the two arch channel beams which are located in the front wall. These have been braced by tieing them to two additional channels, placed back to back and located directly above the channels which gave the trouble. The front arches are open to the air on the outside and air vents have been provided in the setting so that the rear arch is also cooled by natural air currents.

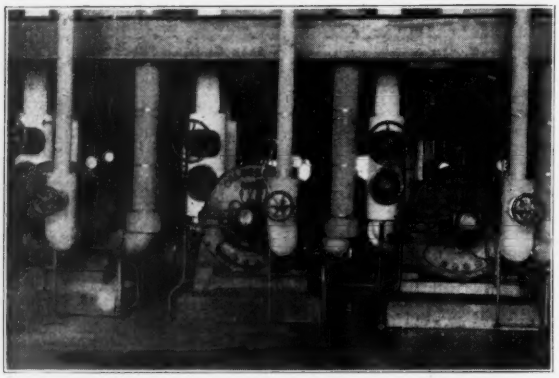


FIG. 3. BOILER FEED PUMPS ARE STEAM TURBINE DRIVEN

As has been previously stated, eight boilers have been installed. These are of the Edge Moor longitudinal type, each being rated at 1020 hp. They are four-pass boilers set with a rear gas passage which leads through the boiler room floor to the breeching. The bottom of the front headers are set about 16 ft. above the floor. These boilers are operated at about 180 lb. pressure and are equipped with Foster superheaters, which superheat the steam to 150 deg. F.

Although no CO_2 recorders have been installed, each setting is completely fitted so that gas samples may readily be taken. This is provided for by a system of ¼-in. piping which is so arranged with crosses and plugs that it can be thoroughly cleaned out should it become plugged with soot or other foreign matter. CO_2 samples are frequently taken and analyzed. Space remains within the present boiler room for the installation of two additional boiler units, one on each side of the firing aisle.

Accessories on each boiler are as follows; consolidated safety valves on both the boiler and superheater, Homestead combination blowoff valves, Edward Valve & Mfg. Co. non-return and feedwater valves, and eight units of diamond Power Specialty Corp. soot blowers.

Expansion Provided for in Breeching

Connery expansion joints are used on the ¼-in. steel plate, unlined breeching, which is located beneath the operating floor. The breeching from two boilers come together into a common breeching, which passes through the wall and enters the side of the stack. One stack takes care of four boilers, a breeching entering from either side. The stacks were constructed by the Heine Chimney Co. of radial brick and they are 255 ft. high by 16 ft. inside diameter at the top. No induced draft fans are used. The

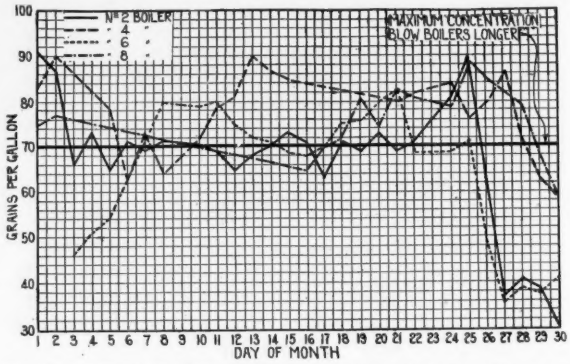


FIG. 5. AVERAGE CONCENTRATION OF THE WATER IN EACH BOILER IS DETERMINED AND THE BOILER BLOWN DOWN ACCORDINGLY

character of the coal used is such that a large amount of fine ash collects in the base of the stacks and to remove this ash, a home-made screw conveyor is used. It is built in two sections so as to form a right angle, the lower section being inserted through the cleanout door in the base of the stack and the doors sealed with the conveyor in place so that the operation of ash removal does not impair

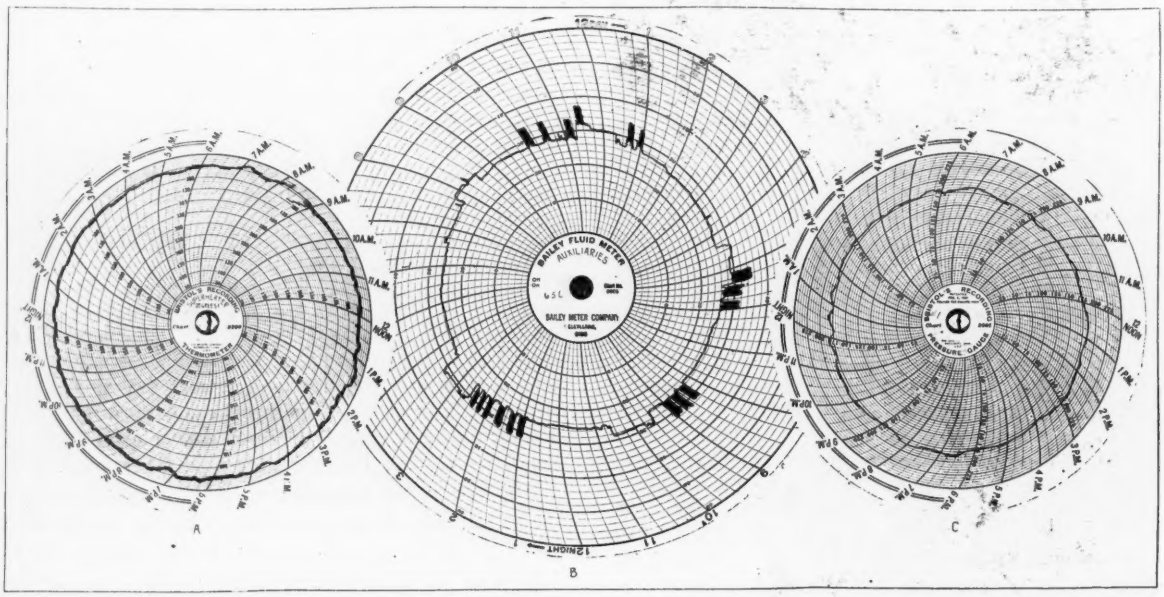


FIG. 4. REGULATION OF THIS PLANT IS ENTIRELY BY HAND. CHARTS "A" AND "C" ARE OF THE STEAM TEMPERATURE AND PRESSURE, RESPECTIVELY AND CHART "B" SHOWS STEAM FLOW TO AUXILIARIES, NOTE THE PERIODIC STEAM DEMAND BY THE SOOT BLOWERS.

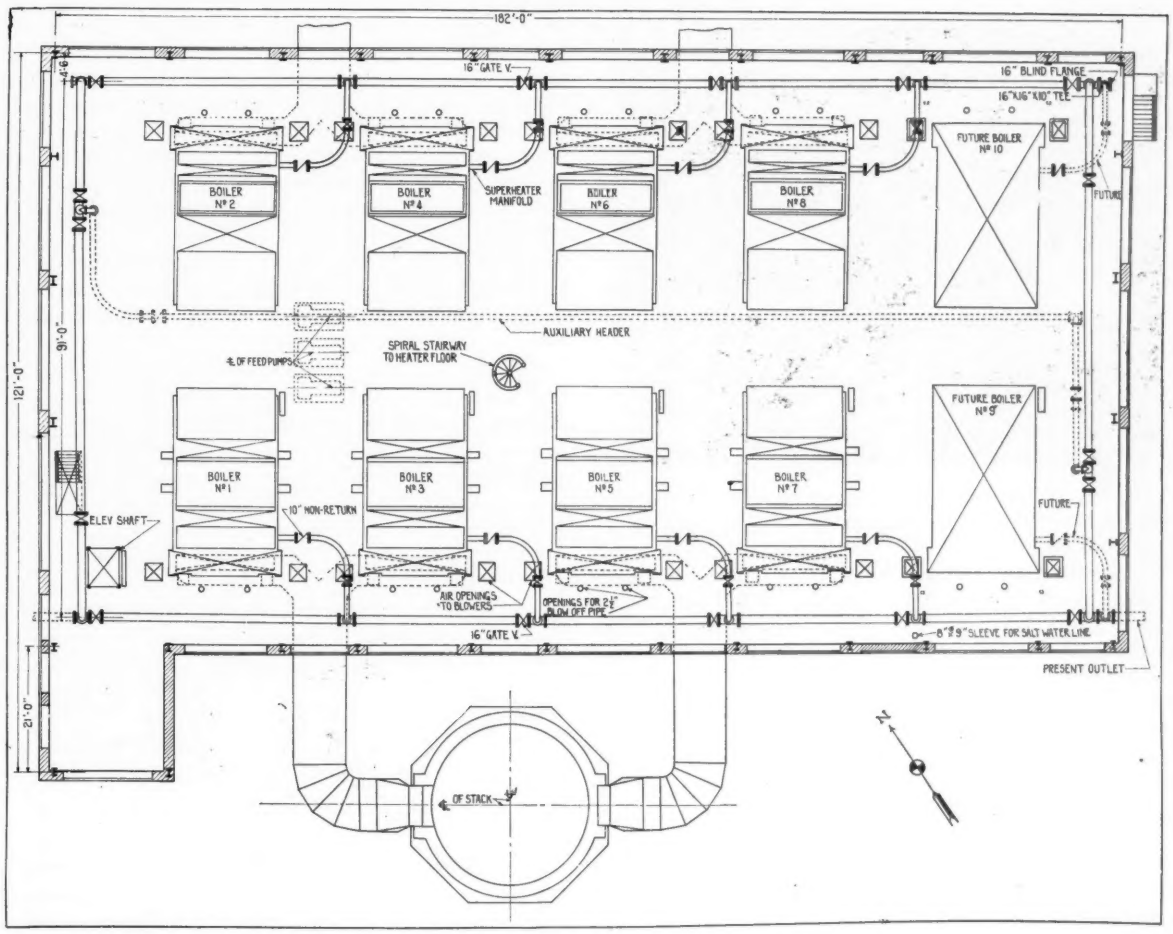


FIG. 6. BOILERS ARE ARRANGED WITH A CENTRAL FIRING AISLE

the effectiveness of the stack. The screw conveyor discharges onto a portable conveyor by means of which wagons or trucks may be loaded with the ash. This ash found in the base of the stacks contains about 35 per cent combustible, no part of which is volatile matter.

Each hopper beneath the stokers is divided into a front and rear compartment by means of a vertical partition. The advantage of this is that the front compartment is used to catch the siftings from the stokers, whereas the rear compartment is used to catch the ash discharged by the stoker. This ash hopper is connected by means of pipes to the space beneath the second and third boiler passes. In this way the dust collected at that point is readily removed. A damper is provided in each dust chute and each chute is capped with a ¾-in. wire mesh hood, which is conical in form, which prevents pieces of baffle and tile from clogging the chutes.

Ash Dumped Directly Into Railroad Cars

The basement floor is level with the yard so that two standard gage tracks enter the building, one being under each row of boilers. The head room is ample so that standard railroad hopper bottom cars can be spotted directly under the Allan-Sherman-Hoff ash hoppers which are provided with water sealed ash gates made by the same company. Some trouble was experienced at first due to switchmen damaging the rolling doors at the entrance-way of the ash track to the building. To call their attention to the position of the door, an electric signal light has been placed near the bottom and on one side of each roller door. This red light burns all the time no matter what the position of the door may be and since its installation the doors have not been damaged by moving cars.

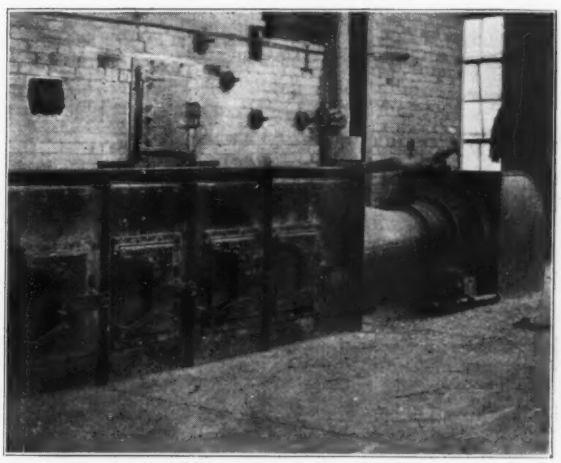


FIG. 7. EACH STOKER IS SERVED BY TWO BLAST FANS, ONE PLACED AT EACH SIDE OF THE SETTING

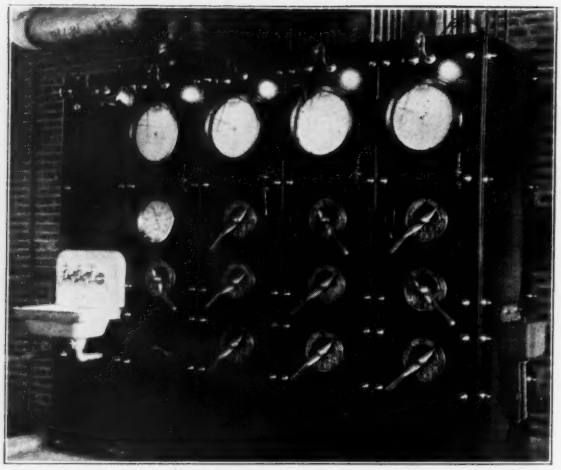


FIG. 8. WATER TREATING PLANT IS EQUIPPED WITH A CENTRAL CONTROL BOARD

High pressure steam piping is laid out in the loop system, each superheater manifold being connected through long radius bends to a 16-in. header running back of the boilers. The ends of these headers are tied together by 16-in. crossovers from which 10-in. connections are made for the auxiliary header which thus also forms a loop below the operating floor. Valves are so arranged that the entire system is flexible. The 16-in. yard lines are connected directly to the east and west ends of the south header. Piping and pipe covering in the boiler house were furnished and erected by the J. M. Dougherty Co. The pipe joints are all Vanstoned and are fitted with Goetze corrugated copper gaskets which are filled with asbestos packing. The valves are all designed for 250 lb. pressure and 750 deg. F. temperature. They are all gate valves with bypass, some having been furnished by Crane Co. and some by the Edward Valve & Mfg. Co.

Feed water is obtained from the city of Bayonne. This water is first used to cool the wax in one of the processes in the refinery. From this process the water is pumped to a large steel storage tank which is located near the west end of the boiler house. Located just south of the boiler plant is a substantial building of reinforced concrete and brick, which houses a Scaife water treating plant which is of the intermittent type and treats the water cold. Two steam driven Alberger-Curtis centrifugal pumps, each rated at 3250 g.p.m., take water from the storage tank and deliver it to the treating plant, where it is treated with lime, soda ash and sulphate of iron. From this treating plant the water is delivered by two 1200 g.p.m. Alberger-Curtis centrifugal pumps to the feed-water heaters, which are located on a steel and concrete platform just below the boiler operating floor. All of the pumps just mentioned are in the treating plant and are located below the grade level. The steam used to drive these pumps comes from a line which dead ends in the water softener plant. The sumps in this plant are below the sewer grade, therefore, a steam ejector serves the double purpose of keeping the steam line free of condensate and also to drain the sump pits.

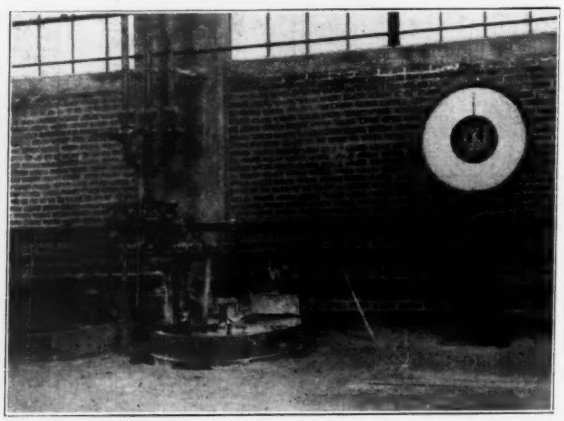


FIG. 9. MIXING TANKS AND SCALES ARE CONVENIENTLY ARRANGED IN THE WATER TREATING PLANT

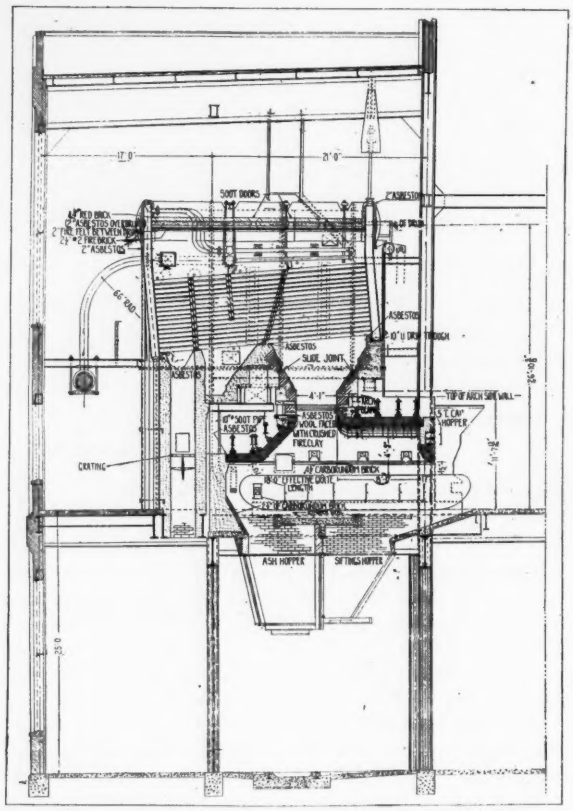


FIG. 10. FURNACES ARE BUILT HIGH AND WITH A COMPARATIVELY NARROW THROAT BETWEEN THE FRONT AND BACK ARCHES

Auxiliary Exhaust Steam Not Sufficient to Heat Feed Water

Two Cochrane cylindrical, horizontal, open-type heaters are in operation, each being rated at 9000 hp., and the water is now being heated to 145 deg. F. by what steam is available from the auxiliaries. An exhaust steam line is being brought in from the yard, which will supply enough low-pressure steam to bring the feed-water temperature up to about the boiling point. Three Alberger-Curtis boiler feed pumps are located directly below the heaters. These are three-stage centrifugal pumps, each of which is rated at 750 g.p.m. and each is fitted so that it can be primed with city water should it be necessary. Two feed lines branch out from the pumps, one being equipped with a Stets

feed-water regulator at each boiler and the other being valved for hand control. A check valve is in the discharge line of each pump. All high-pressure water lines are constructed with screwed flanges, which are sealed with Klingerit gaskets. All valves were furnished by Crane Co. and are of the gate type. The hot water lines are covered with a 3-in. covering furnished by Johns-Manville, Inc.

Boilers are blown down, not on a schedule of so many hours but in accordance with the concentration of the water within the boiler. A number of analyses are made each day of the boiler water concentration and the results are plotted on a chart. It is aimed to keep the concentration below 70 gr. per U. S. gallon, this point being indicated on the chart. If the concentration goes above this for any boiler, it is blown down more often; if the concentration is below 70, the boiler need not be blown down so often. The blowoff tank for this plant is of steel and is mounted on a platform outside of the building and some distance above the ground level.

No great complexity of instruments has been installed in this plant. Two Venturi meters, made by the Builders Iron Foundry, are used to measure the feed water, one being on the line which is regulated and the other on the feed line which is hand controlled. The recorder mechanisms are mounted in the engineer's office. The office is also equipped with Bristol recorders for feed-water pressure in each line and the temperature of the water to the boilers and steam pressure. A Bailey flow meter has been installed, which will be used for test purposes only.

Boilers Are Equipped With Individual Instrument Boards

At each boiler is placed an instrument board on which are mounted the following instruments: a Bailey boiler meter, a Bristol recorder giving the temperature of the steam, a Bristol steam pressure recorder, a U. S. Gauge Co. steam pressure indicator and a Bailey 8-point draft and air pressure indicator. In the near future two 12-in. feed-water pressure indicating gages will be installed so they can be seen from any point in the firing aisle.

When operating at about 170 per cent of rating, the air pressures in inches of water on an Illinois stoker were as follows:

1st zone pressure	plus 0.10
2nd zone pressure	plus 0.50
3rd zone pressure	plus 0.80
4th zone pressure	plus 1.10
5th zone pressure	plus 1.20
6th zone pressure	plus 1.60
7th zone pressure	plus 1.40
8th zone pressure	plus 0.10
Draft in firebox	minus 0.05
Draft at damper	minus 1.20

Similar figures on a Harrington stoker were as follows:

1st zone pressure	plus 0.40
2nd zone pressure	plus 1.00
3rd zone pressure	plus 1.00
4th zone pressure	plus 1.00
5th zone pressure	plus 0.70
6th zone pressure	plus 0.10
Draft in firebox	minus 0.05
Draft at damper	minus 1.10

One Bailey flow meter is installed on each outgoing steam line and the auxiliary header is also metered. An interesting feature in connection with the auxiliary header is that the soot blowers take steam after it has passed through the meter, thus, the meter chart shows each time a soot blower was used and for how long a period. Connections are also made so that the soot blowers can also take steam from a connection near the superheaters.

Maintenance Machine Shop Will Be Installed

In the east end of the center bay, on the ground floor, will be located a small machine shop which will be used in connection with the maintenance of the boiler plant. On this floor are also located a Worthington steam driven air compressor used to supply air for tools and also two 15 by 10½ by 18-in. duplex pumps which handle salt water for general purposes, such as washing boilers. Fresh water is used for all sanitary purposes. One other feature of the water piping is that the feed pumps are so piped that they can take cold raw water, heated raw water, treated cold water, or treated hot water.

At the present time, the shifts are divided into eight hours each; one engineer, two firemen, a man on the weigh larry and one man in the water treating plant are on each shift. Two men are required by the coal handling equipment for about 5 hr. out of each day. Laborers take care of the ash and dumping coal on the day shift. It is probable that as the load increases that one additional man will be added to each shift.

This plant stands on filled ground and the foundations rest on piling. The building is of steel and brick and the roof is of gypsum tile overlaid with tar and gravel. The floors are of concrete and steel and the platforms and stair treads are of diamond steel plate, all hand rail being made of angle iron. An interesting feature of the side wall construction at the operating floor level is the great number of roller doors provided for the purpose of ventilating the room during hot weather. The engineer's office is of generous proportions and finished off with white tile to a height of about 5 ft. The wash and shower room is ventilated by an electric fan. For the convenience of the operators there has been installed an Otis automatic elevator which operates between the ground or lower level and the roof of the building.

Line Moved Without Interruption

The juggler who tosses blazing oil lamps about as if they were Indian clubs may be an old story, but we cannot see such a performance without marveling at the dexterity displayed. Such a feat is, however, after all simply a matter of practice, but recently, as just part of the day's work in Ohio six miles of electric light tension transmission line carrying electricity at 13,000 v. was transplanted without cutting off the flow of current. The transmission line paralleled a road leading from Martinsville to New Vienna which was to be widened to 60 ft. This required that the electric line poles be moved back 10 ft.

A crane mounted upon a big truck was sent out and one by one it picked up the poles supporting the wires of this 13,000-v. line and placed then gently in freshly dug holes 10 ft. farther back from the highway. Sometimes the crane moved these poles at the rate of one every 20 min., but on an average only 16 poles, or about 0.6 mi., of the transmission line could be transplanted in a single day. The entire job was done without accident or interruption of electrical service.

Furnace Design for Burning Anthracite

Abstract of Paper Presented at the A. S. M. E. Meeting in New York, Dec. 1, 1924. By W. A. Shoudy and R. C. Denny

ANTHRACITE, on account of its low volatile and high carbon content, is used extensively in domestic furnaces, but in the process of crushing it for this purpose, smaller sizes are produced which are not in demand for household use. These sizes must be sold or the entire burden will be placed on users of domestic sizes.

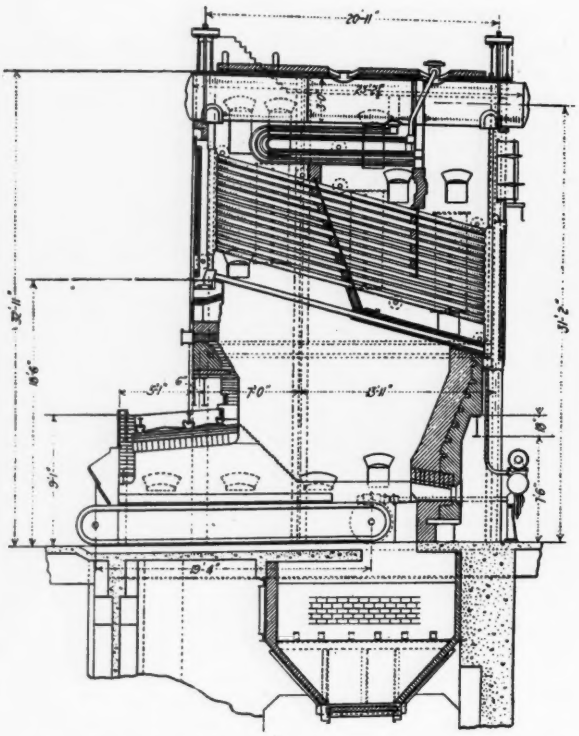


FIG. 1. SINGLE ARCH FURNACE FOR BURNING ANTHRACITE. ORIGINAL SETTING OF BOILERS NO. 1 AND NO. 2

Pea, No. 1 and No. 2 buckwheat are used to some extent in heating boilers and in large boilers where firemen are necessary. There still remains a size referred to here as No. 3 buckwheat that can be used only for power requirements. This size the writers have designated as coal of which "not less than 95 per cent will pass through a $\frac{3}{16}$ in. round mesh screen and not more than 20 per cent through a $\frac{3}{32}$ in. round-mesh screen." (Coal smaller than the $\frac{3}{32}$ in. size will be referred to here as undersize.) It is for methods of burning this size between $\frac{3}{16}$ in. and $\frac{3}{32}$ in. that the furnace constructions described here have been developed.

There are in general only two satisfactory methods of burning this coal; i.e., by hand firing and by chain grate stokers. While progress in using it as pulverized fuel has not been rapid, the process promises a means of disposing of the "undersize."

Until a few years ago, the ignition difficulties inherent in anthracite have made it necessary to shield the ignition end of the furnace from the cooling effect of the boiler tubes by a refractory arch, absolutely necessary with a chain-grate stoker and useful with hand fires. This type of furnace, illustrated in Fig. 1, with large combustion volume and grate extending under the full length of the furnace, will give as high an efficiency as can be obtained

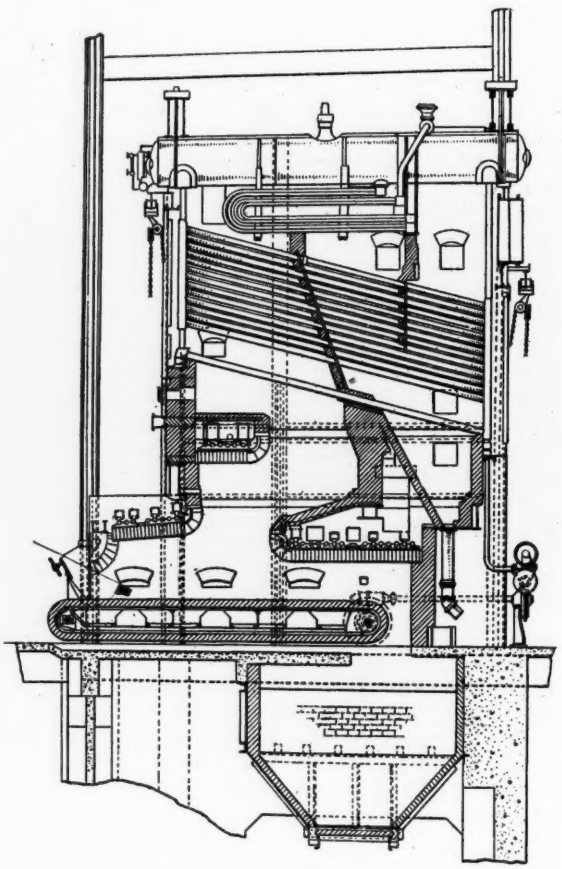


FIG. 2. TRIPLE-ARCH EXPERIMENTAL FURNACE (BOILER NO. 1) AT AMSTERDAM

with a single arch. It has been accepted as standard design, the low efficiencies as compared with bituminous coal being accepted as inevitable.

Objections to this type of furnace are:

a. Tendency to stratification of gases
b. High carbon content of the ash
c. Loss of fires to ashpit and back connections
d. Lack of flexibility.

On a chain grate, combustion is progressive. At the front end is generally found high CO_2, in the center high CO and at the rear end considerable excess air. For complete combustion, the latter is necessary and in this type of furnace the excess air generally follows a well-defined lane throughout the setting. In order to reduce stack loss, the operator must either reduce excess air, thus increasing carbon loss to the ashpit, or else reduce carbon loss at the expense of stack loss. Lack of uniformity in sizing the coal leads to a closing up of air spaces in the fuel bed; this necessitates a higher air pressure beneath the grate that blows particles of coal onto the tubes, back connection

and breeching, causing a direct loss. Finally, the length of the fire varies with the load; the operator must, therefore, carry a long fire and waste coal at light loads or take his chances of maintaining steam pressure with shorter fires. These, in brief, are some of the disadvantages of the type of furnace shown in Fig. 1.

For several years the subject has been studied. At the Baltimore refinery of the American Sugar Refining Co., boilers were installed, to burn anthracite, in which a second arch was built over the rear of the grate. The authors' experiences with those furnaces convinced them that a distinct advance had been made.

In 1921, the Amsterdam Steam Station of the Adirondack Power & Light Corp. was started, using a 15,000 kw. turbine and two 1345-hp. boilers. The latter are of B. & W. design, 42 tubes wide, 14 tubes high with 20 ft. tubes. Each boiler is set with two Coxe traveling-grate stokers 10 ft. 8⅝ in. wide by 17 ft. long, for burning anthracite. The load is extremely variable, as the station is a relay to a hydro-electric system; for this reason, difficulty was experienced in operating the boilers. Installation of another turbine and two more boilers caused a little improvement, the second group of boilers being set with short rear arches.

Finally boiler No. 1 was equipped with an experimental furnace as shown in Fig. 2. An extended series of tests was then made of this furnace and of boiler No. 2, an identical boiler, but equipped with the old furnace of Fig. 1. The results of these tests are plotted in broken lines. In Fig. 3, only three tests being run on boiler No. 2 because its characteristics were so well known.

From a combustion standpoint, the new furnace proved more than satisfactory, but an operating difficulty made it necessary to redesign it. The one trouble that developed was the accumulation of a mixture of carbon and slag, deposited in plastic form on the rear arch so thickly that the boiler had to be taken out for cleaning after about three weeks of continuous operation.

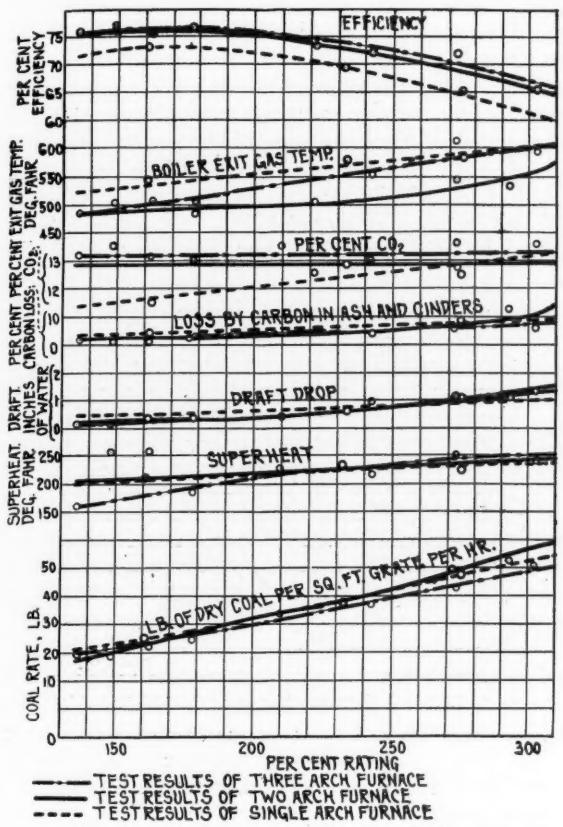

EFFICIENCY
PER CENT EFFICIENCY
BOILER EXIT GAS TEMP.
EXIT GAS TEMP. DEG. FAHR
PER CENT CO₂
PER CENT CO₂
LOSS BY CARBON IN ASH AND CINDERS
PER CENT CARBON LOSS
DRAFT DROP
DRAFT, INCHES OF WATER
SUPERHEAT
SUPERHEAT DEG. FAHR
LB. OF DRY COAL PER SQ. FT. GRATE PER HR.
COAL RATE, LB.
PER CENT RATING
TEST RESULTS OF THREE ARCH FURNACE
TEST RESULTS OF TWO ARCH FURNACE
TEST RESULTS OF SINGLE ARCH FURNACE

FIG. 3. CHART SHOWING TEST RESULTS OF THREE TYPES OF FURNACES

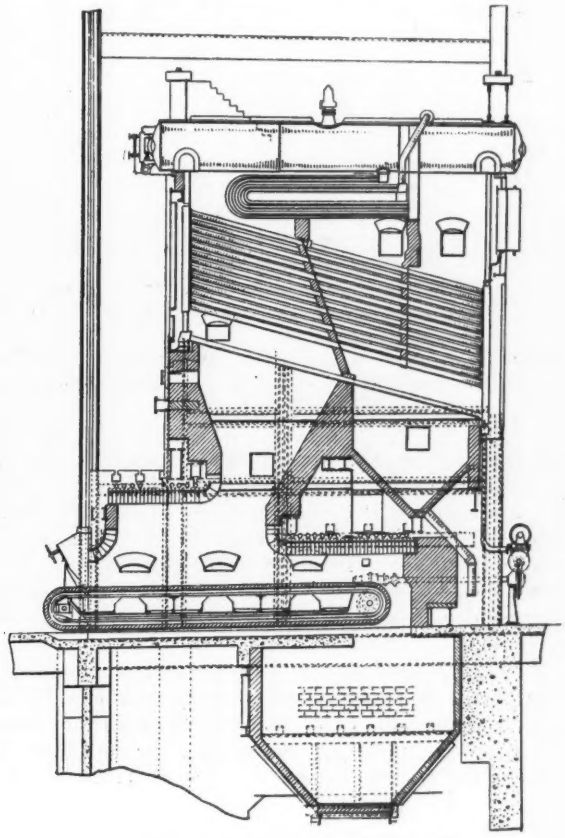


FIG. 4. TWO-ARCH DESIGN WITH RESTRICTED GAS PASSAGE (FINAL DESIGN OF BOILER NO. 1)

It was impossible to rebuild the furnace immediately, since there was no spare boiler equipment in the station. During the next run of the boiler, however, observations were made of the periodic building up of the slag. It was found that there was little increase in draft drop with the reduction in gas passage; accordingly, the slag was allowed to accumulate until the opening between the upper arch and the slag was reduced to about a foot. Then the boiler was operated at 250 per cent of rating, showing an increased drop of only 0.2 in. of water.

Since this experience indicated the possibility of securing complete mixture of gases by using a restricted throat, boiler No. 2 was reset as shown in Fig. 4. Six tests of this boiler, plotted in Fig. 3, show that, at ratings below 250 per cent, the efficiencies check closely with the tests of the three-arch furnace, the two tests at the higher ratings showing a marked falling off in efficiency. The latter two tests were made at a period when it was difficult to regulate the load, as there were only two boilers in service, one under test and one under light load to

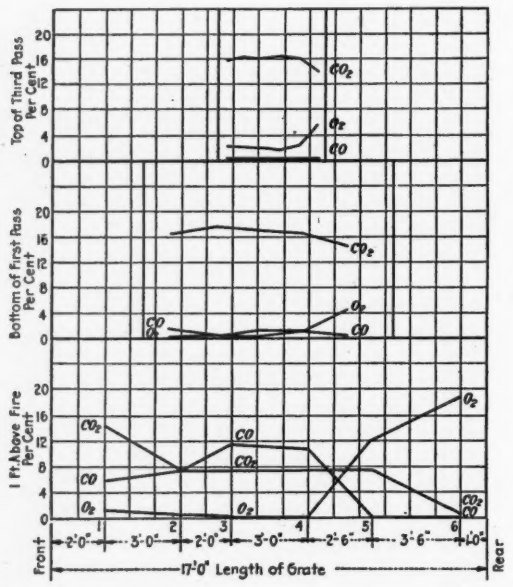


FIG. 5. VARIATIONS OF FLUE GAS COMPOSITION IN THREE-ARCH FURNACE. (SPECIAL RUN, AUG. 21, 1923, 275 PER CENT RATING)

take the irregularities. Below 170 per cent of rating it was difficult to regulate conditions because of the low rate of combustion. The two tests below this rating show high efficiency, a high unaccounted-for loss, high superheat and high exit gas temperatures. Only traces of CO were found, but undoubtedly some secondary combustion occurred due to stratification caused by low velocity of the gases.

Efficiency curves for the three furnaces, super-imposed as in Fig. 3, show clearly the improvement obtained. The point of maximum efficiency for all three furnaces is about 170 per cent of rating, but the curves of the multiple-arch furnaces are flatter. At this rating the increase in efficiency is slightly over three points, rising to about five points at 300 per cent rating.

Reasons for this improvement in efficiency are apparent from a study of the analyses of flue gas taken on the three-arch furnace and plotted in Figs. 5, 6 and 7. Fig. 5 shows the influence on combustion of the three arches. Samples were taken through a water-cooled sampling tube 12 in. over the fire and about 30 in. from the side wall, with the boiler operating at about 275 per cent of rating. The lower set of curves shows the points at which samples, representing products of combustion of the various air compartments of the stoker, were drawn. Some air is used in the first five compartments at ratings above 200 per cent, but none is admitted to the sixth. Air was admitted to burn the fire in what seemed the best way and not to secure any particular arrangement for draft gage readings. A second set of five analyses was made in the bottom of the first pass, using the water-cooled sampling tube introduced between the second and third rows of tubes from the bottom, since that was the only way in which the sampling pipe could be introduced. As soon as these analyses were finished, six samples were drawn from the top of the third pass through the regular gas-sampling tubes.

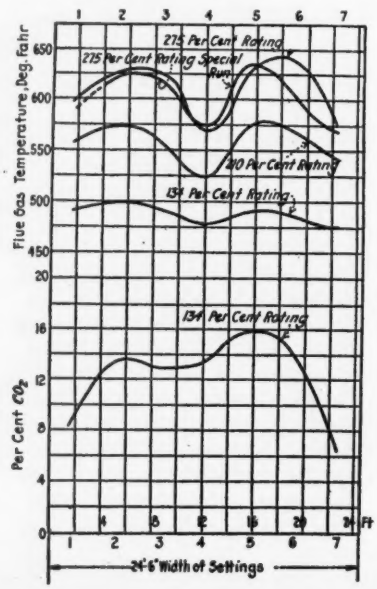

Flue Gas Temperature, Deg. Fahr
275 Per Cent Rating
275 Per Cent Rating Special Run
210 Per Cent Rating
134 Per Cent Rating
134 Per Cent Rating
Per Cent CO_2
24'-6" Width of Settings

FIG. 6. TEMPERATURE AND CO_2 ACROSS TOP OF THIRD PASS OF THREE-ARCH FURNACE

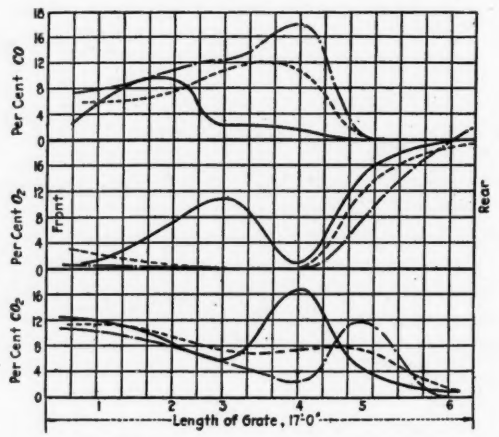


FIG. 7. VARIATIONS IN FLUE GAS COMPOSITION OVER FIRE IN THREE-ARCH FURNACE

Full lines—274 per cent rating. Aug. 17, 1923.
Dotted lines—230 per cent rating. Aug. 21, 1923.
Dot-and-dash lines—134 per cent rating. Aug. 19, 1923.

These three sets of readings are in almost the same vertical plane throughout the setting, at similar firing conditions. Only a moderate amount of CO, about 6 per cent, is produced over the first compartment, the CO_2 at the same point being about 14 per cent. Over the third compartment the CO increases to nearly 12 per cent, while the CO_2 decreases to 7 per cent until the fifth compartment is reached. Here excess air is admitted to burn out the fuel bed, much less carbon being burned here in proportion to the air admitted through the fire. CO_2 does not increase, but the free O rises to 12 per cent and the CO disappears. Over the sixth compartment neither CO or CO_2 occurs in any appreciable quantity.

Inspection of the curves of Fig. 5 in the light of the foregoing analyses shows that combustion is practically complete at the bottom of the first pass, the CO being all burned out and little free O being present except in the section nearest the baffle. The presence of

TABLE OF DATA AND RESULTS OF EVAPORATIVE TESTS

Test number		1	2	3	4	5	6	7	8	9	21	22	23	24	26	27
Boiler number		1	1	1	1	1	1	2	2	2	2	2	2	2	2	2
Grate surface,	sq. ft.	364	364	364	364	364	364	364	364	364	364	364	364	364	364	364
Water heating surface,	sq. ft.	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450	13,450
Superheating surface,	sq. ft.	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863	3,863
Total heating surface,	sq. ft.	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313	17,313
Ratio water heating to grate surf.		37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1	37-1
Ratio total heating to grate surf.		47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1	47.6-1
Volume of combustion chamber,	cu. ft.	2,690	2,690	2,690	2,690	2,690	2,690	6,140	6,140	6,140	.....	.....	.....	.....	.....	.....
Date		9/24/23	8/17/23	9/25/23	8/17–18/23	9/26/23	8/17–19/23	8/18–24/23	8/25/23	26/23	8/25–9/23/24	9/3/24	9/4/24	9/5/24	11/11/24	11/12/24
Duration,	hr.	8	8	8	9	8	9	8	8	8	8	8	8	8	8	8
Coal (anthracite):																
Oversize,	per cent	13.2	26.32	13.3	22.22	13.6	25.0	25.7	32.06	31.70	17.6	19.0	14.7	13.6	14.8	14.6
No. 3 Buckwheat,	per cent	51.2	57.87	50.5	59.28	59.3	60.0	48.6	41.52	46.34	48.5	49.7	48.0	46.0	52.0	52.6
Undersize,	per cent	35.6	15.81	36.2	18.50	27.1	15.0	25.7	26.42	21.96	33.9	31.3	37.3	40.4	33.2	32.8
Steam pressure by gage,	lb. per sq. in.	272.5	279.8	265.9	272.0	274.1	285.4	267.0	271.7	271.7	286.0	281.1	285.6	284.6	273	272
Barometer,	in. mercury	30.0	29.9	30.0	29.9	30.0	29.8	29.9	29.8	29.9	29.9	29.9	29.8	29.5	30.0	30.0
Steam temperature,	deg. fahr.	660	672	637	638	604	577	641	641	620	650	629	605	602	669	669
Saturated steam temp.,	deg. fahr.	413.6	415.8	411.5	413.4	414.4	417.5	411.8	412.9	412.9	417.8	416.2	417.8	417.8	414	413
Boiler-feed temperature,	deg. fahr.	164.0	146.4	154.0	156.3	152.0	156.3	141.0	146.0	145.0	185.0	191.0	190.0	197.0	180	179
Flue-gas temperature,	deg. fahr.	593	616	555	557	508	488	585	572	536	548	537	505	490	506	512
Draft betw. damper and boiler,	in. H_2O	1.30	1.16	0.90	0.38	.....	0.09	0.97	0.67	0.48	1.13	0.94	0.80	0.79	0.255	0.194
Draft in main flue,	in. H_2O	2.61	1.60	1.24	1.05	0.32	0.50	1.80	1.42	1.72	1.50	0.95	1.37	1.42	0.370	0.470
Draft in furnace,	in. H_2O	0.00	−0.02	−0.01	0.00	+0.03	0.00	+0.01	−0.01	0.00	0.00	−0.026	−0.025	−0.135	+0.009	+0.020
External air temperature,	deg. fahr.	70	68	69	62	67	65	58	63	55	58	64	60	60	40	39
Entering air temperature	deg. fahr.	88	86	82	75	79	91	82	85	78	83	82	82	80	40	39
Superheat,	deg. fahr.	264.4	256.2	225.5	224.6	189.6	159.5	229.2	228.1	207.1	232.2	212.8	187.2	184.2	255	256
Total coal as fired,	lb.	148,021	129,227	108,000	107,736	74,022	62,895	137,624	111,341	72,404	154,489	144,796	107,409	82,738	69,572	63,046
Moisture as fired,	per cent	8.51	11.85	9.40	11.24	9.69	10.15	10.26	10.97	10.68	9.55	10.20	9.83	10.44	9.04	9.62
Total dry coal,	lb.	135,400	113,900	97,900	95,600	66,900	56,500	123,400	99,100	64,700	139,800	130,000	96,800	74,200	63,286	56,990
Ash from ashpit,	lb.	30,750	25,900	20,100	21,470	13,420	8,530	30,900	20,900	11,800	39,500	30,600	21,800	15,700	12,160	10,240
Cinders caught,	lb.	2,096	1,802	1,072	445	223	191	.....	.....	.....	.....	.....	.....	.....	.....	.....
Total refuse,	lb.	32,846	27,702	21,172	21,915	13,643	8,721	30,900	20,900	11,800	39,500	30,600	21,800	15,700	12,160	10,240
Total combustible burned,	lb.	102,554	86,198	76,728	73,685	53,257	47,779	92,500	78,200	52,900	100,300	99,400	75,000	58,500	57,412	52,806
Total refuse based on dry coal,	per cent	24.2	24.3	21.6	22.9	20.4	15.4	25.0	21.1	18.2	28.2	23.5	22.5	21.2	19.2	18.0
Total water fed,	lb.	889,610	798,300	722,115	703,605	525,470	461,555	772,930	685,315	481,305	903,394	844,892	722,237	555,660	480,084	436,984
Factor of evaporation		1.253	1.277	1.251	1.249	1.252	1.213	1.267	1.261	1.246	1.225	1.205	1.194	1.185	1.242	1.243
Total equiv. water fed,	lb	1,115,000	1,020,000	903,500	879,000	658,000	560,000	979,500	864,000	599,500	1,107,000	1,018,000	862,000	658,000	596,000	544,000
Dry coal per hour,	lb.	16,940	14,250	12,230	10,610	8,355	6,280	15,400	12,400	8,090	17,470	16,250	12,105	9,260	7,911	7,124
Dry coal per sq. ft. grate per hr.,	lb.	46.5	39.1	33.6	29.1	23.0	17.3	42.3	34.1	22.2	48.0	44.7	33.3	25.4	21.7	19.6
Water evap. per hr. corr. for quality,	lb.	111,201	997,888	90,264	78,178	65,684	51,284	96,616	85,664	60,163	112,924	105,612	90,279	69,458	60,011	54,623
Equiv. evap. per hour,	lb.	139,400	127,500	112,850	97,660	82,280	62,220	122,500	108,100	74,950	137,200	127,250	107,800	82,300	74,534	67,896
Equiv. evap. per hr. per sq. ft. h. s.,	lb.	10.37	9.48	8.39	7.26	6.12	4.63	9.11	8.04	5.57	10.2	9.47	8.02	6.12	5.55	5.06
Equiv. evaporation per hour,	lb.	139,400	127,500	112,850	97,660	82,280	62,220	114,500	108,100	749,500	137,200	127,250	107,800	82,300	74,534	67,896
Boiler hp. developed,	hp.	4,042	3,695	3,271	2,828	2,385	1,804	3,420	3,135	2,240	3,975	3,690	3,125	2,386	2,160	1,968
Rated boiler hp.	hp.	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345	1,345
Rating developed,	per cent	300.5	275.0	243.1	210.0	177.4	134.0	254.0	233.0	161.0	295.0	274.0	232.0	177.0	161	147
Water fed per lb. coal as fired,	lb.	6.01	6.18	6.69	6.53	7.10	7.34	5.25	6.15	6.65	5.85	5.84	6.72	6.72	6.91	6.93
Water evap. per lb. dry coal,	lb.	6.57	7.01	7.37	7.35	7.86	8.17	5.85	6.91	7.43	6.47	6.49	7.46	7.50	7.58	7.68
Equiv. evap. per lb. coal as fired,	lb.	7.54	7.89	8.36	8.15	8.90	8.90	6.65	7.75	8.28	7.17	7.03	8.02	7.95	8.57	8.63
Equiv. evap. per lb. dry coal,	lb.	8.23	8.94	9.22	9.20	9.85	9.90	7.36	8.72	9.28	7.85	7.83	8.91	8.89	9.42	9.55
Equiv. evap. per lb. combustible,	lb.	10.87	10.62	11.77	11.92	12.35	11.72	9.90	11.04	11.33	11.02	10.2	11.5	11.3	10.38	10.23
Heat value per lb. dry coal (calc.)	B.t.u.	12,370	12,110	12,400	12,155	12,470	12,630	11,932	12,175	12,310	12,260	11,850	11,675	11,265	12,084	12,095
Efficiency—boiler—grate—furnace	per cent	64.6	71.6	72.1	73.5	76.7	76.0	64.5	69.5	73.2	62.2	64.2	74.0	76.6	75.6	76.6
Average fire thickness,	in.	4.4	4.6	5.5	4.8	4.6	3.9	4.5	4.5	4.4	5.0	4.5	4.5	4.5	4.5	4.25
Stoker speed,	ft. per hr.	44.3	36.7	31.7	29.8	21.2	18.9	40.5	33.0	21.6	44.8	41.0	30.3	24.5	20.7	19.7
Flue gas																
Carbon dioxide,	per cent	13.56	12.77	13.08	13.56	13.00	13.14	12.48	12.80	11.58	11.30	13.00	12.50	11.4	13.1	13.5
Oxygen,	per cent	6.10	5.81	6.80	5.73	6.58	6.34	6.34	6.08	7.62	7.67	5.27	7.05	8.0	6.3	6.5
Carbon monoxide,	per cent	0.00	0.02	0.00	0.23	0.00	0.06	0.23	0.45	0.07	0.69	1.23	0.00	0.0	0.16	0.0
Nitrogen,	per cent	80.34	81.40	80.12	80.48	80.42	80.46	80.95	80.67	80.73	80.34	79.90	80.45	80.6	80.44	80.0
Proximate analysis																
Moisture,	per cent	8.51	11.85	9.40	11.24	9.69	10.15	10.26	10.97	10.68	9.55	10.20	9.83	10.44	9.04	9.62
Volatile matter,	per cent	8.07	6.22	6.58	6.67	5.98	5.62	6.20	6.18	6.17	8.16	7.44	6.93	5.91	7.36	7.30
Fixed carbon,	per cent	65.77	66.26	67.55	66.44	68.48	71.95	67.02	67.62	69.97	66.63	65.91	67.20	66.68	67.04	67.04
Ash,	per cent	17.65	15.67	16.47	16.65	15.85	12.28	16.52	15.23	13.18	15.66	16.45	16.04	16.97	16.56	15.44
Sulphur (dry basis),	per cent	0.73	1.24	0.54	1.05	0.34	0.75	0.69	0.63	0.72	0.56	0.69	0.59	0.74	0.51	0.61
Ultimate analysis (calculated)																
Carbon,	per cent	73.00	75.52	75.32	74.09	76.55	79.57	74.93	76.11	78.42	75.97	75.05	75.58	74.59	74.86	75.92
Hydrogen,	per cent	2.99	2.55	2.73	2.36	2.61	2.44	2.54	2.52	2.47	2.76	2.97	2.76	2.60	2.74	2.79
Oxygen,	per cent	3.27	2.08	2.73	2.35	2.49	2.73	2.60	2.79	2.78	2.88	2.07	2.38	2.65	2.79	2.69
Nitrogen,	per cent	0.62	0.84	0.51	0.80	0.46	0.84	0.84	0.84	0.85	0.51	0.92	0.89	0.46	0.88	0.89
Sulphur	per cent	0.73	1.24	0.54	1.05	0.34	0.75	0.69	0.63	0.72	0.56	0.69	0.59	0.74	0.51	0.61
Ash,	per cent	19.30	17.77	18.17	18.75	17.55	13.67	18.40	17.11	14.76	17.32	18.30	17.80	18.96	18.22	17.10
Ash analysis																
Volatile matter + carbon,	per cent	15.00	21.85	11.45	16.50	12.50	9.55	26.40	18.85	19.15	32.30	22.25	21.00	10.53	5.10	4.82
Earthy matter,	per cent	85.00	78.15	88.55	83.50	87.50	90.45	73.60	81.15	80.85	67.70	77.75	79.00	89.47	94.90	95.18
Heat Balance—Dry Basis																
Heat absorbed by boiler,	per cent	64.60	71.60	72.10	73.50	76.70	76.00	64.5	69.50	73.20	62.20	64.20	74.00	76.60	75.6	76.6
Loss due to moisture in coal,	per cent	0.88	1.30	0.96	1.20	0.97	0.70	1.10	1.10	1.10	1.10	1.20	1.20	1.30	1.1	1.1
Loss due to hydrogen burned,	per cent	2.77	2.50	2.50	2.20	2.34	2.10	2.40	2.40	2.30	2.60	2.80	2.60	2.60	2.6	2.7
Loss due to flue gases,	per cent	12.55	14.00	12.74	12.30	11.84	11.30	13.70	13.20	14.50	13.00	11.00	12.50	14.00	8.3	8.4
Loss due to carbon monoxide,	per cent	0.00	0.10	0.00	1.20	0.00	0.30	0.90	1.80	0.30	3.20	5.00	0.00	0.00	0.8	0.0
Loss due to combustible in ash,	per cent	4.01	6.00	2.77	4.40	2.96	1.70	*8.10	*4.80	*4.20	*9.80	*6.50	*5.90	*2.90	*1.2	*1.1
Loss due to carbon in cinders,	per cent	1.60	1.70	1.13	0.50	0.35	0.30	.....	.....	.....	.....	.....	.....	.....	.....	.....
Unaccounted for,	per cent	13.59	2.80	7.64	4.70	4.84	7.60	9.00	7.20	4.40	8.10	9.30	3.80	2.60	10.4	10.1
Total,	per cent	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00

* Includes cinder loss

oxygen here shows that stratification has apparently followed the tortuous path of gas travel. At nearly all points in the first pass, the CO_2 is above 16 per cent, dropping to about 14 per cent near the baffle. The indications are that the minimum amount of excess air has been used on this longitudinal section of the fire. The curve of gas analysis at the top of the third pass shows substantially the same CO_2, but a uniformly low amount of CO. A higher amount of free oxygen at this point than at the bottom of the first pass leads to the inference that air has leaked into the setting between the two points.

Figure 6 shows a comparison of gas analyses over the fire at three different ratings; the difference between the curves results from a difference in handling the fires. On Aug. 17 somewhat higher air pressures were carried on the second and third air compartments, thus lowering the CO_2, increasing the oxygen, but reducing the CO. The air pressure was a little lower in the fourth compartment

than in the other run and the excess air was reduced almost to zero, while the CO_2 rose abnormally to 16 per cent. At the fifth compartment occurs a characteristic disappearance of CO and a great increase in free oxygen, while over the sixth compartment the results of all runs show the same character of gas.

In making the low-rating run shown by the dot-and-dash line, with much less air pressure under all compartments, the penetration of the air through the fire was less than in the high-rating runs and the curves show exactly what might be expected. All these results are valuable in showing how gas mixing has occurred in the furnace. The analyses is an important guide in the redesign of the furnace as shown in Fig. 1, because it gives a clear idea of how the three-arch furnace produced the results obtained.

"Cross-Sectioning" Exit Gases

It proved feasible to cross-section the area at the top of the third pass with thermo-couples and gas sampling tubes and to get a fair average of seven simultaneous gas samples across the setting; the readings of the thermo-couples could be taken with the potentiometer almost instantaneously. A curve of the analyses of the seven samples obtained at one point in the test is shown in Fig. 7.

It should also be remembered that there are two stokers with a three-foot center wall across the setting. The shape of the curve indicates considerable longitudinal stratification. At the two end pipes, the low CO_2 was frequently remarked, not only on the 134 per cent rating but also at other ratings. If a single gas sampling tube had been extended in from either side, the CO_2 recorded for the entire series of No. 1 boiler tests would have been about 16 per cent. It is evident that such a reading would have been seriously misleading. Furthermore, if flue temperatures are taken at one point only in the setting, the curves on the upper part of Fig. 7 show that there will be the same kind of error as by taking the CO_2 at one point.

Conclusions from the Experiments

These tests, which are presented not as representing the ultimate development of this type of furnace but as pointing to future development, show clearly that stratification can be eliminated and carbon loss to the ashpit reduced to a minimum. Even with low grade coal, ignition troubles can be overcome, although there is room for improvement in burning undersize at ratings above 250 per cent. All the tests were made with coal of approximately the same quality, otherwise no just comparison could be made. Operation has been studied to some extent with coal containing as high as 60 per cent of undersize, carrying 300 per cent rating with the two-arch furnace as against 190 per cent rating with the single arch. Estimates of loss in efficiency with this coal indicate that the loss is less with the new type of furnace. The problem of burning this undersize has already become important to both consumer and some operators and the development of these furnaces may help in the ultimate solution.

Automatic Combustion Control

Theoretical Considerations Governing Application to Pulverized Fuel Installations. By S. C. Martin

COMBUSTION control has for its object the regulation of the amount of fuel and air supplied to the furnace in accordance with steam requirements and to maintain at all times the most efficient combustion.

In order to apply automatic regulation successfully, it is desirable that the factors entering into the control should be few in number, and their characteristics subject to little variation.

Conditions favorable to successful automatic combustion control may be more easily attained with pulverized fuel installations than with hand fired or stoker fired furnaces. In the former we avoid trouble from ashes, clinkers, holes in the fuel bed and choking of the draft. There is no necessity for forced draft and infiltration of air through ash doors, fire doors and other openings in the furnace walls is reduced, as it is possible to construct and successfully operate a boiler furnace, fired with pulverized fuel, with no openings in the furnace walls other than those for the burner and the removal of the ash.

The main factors with which we have to deal when applying combustion control to pulverized fuel installations are: Steam pressure, steam flow and CO_2 per cent which are controlled by fuel supply, air supply and uptake draft.

It is desirable to maintain the following conditions at all times:

1st. Uniformly fine pulverization.
2nd. Uniform grade of fuel.
3rd. Uniform air pressure at the air regulating valve.
4th. Uniform and reliable action of the fuel feeder.

Without such uniformity no automatic regulation can be expected to operate successfully.

Furnace construction should be given special attention. Air pockets, gas pockets, and waste space should be avoided. Excess air ports should be omitted. Furnace walls should conform as closely as possible to the contour of the flame and the travel of the furnace gases. The burner should be of simple construction and designed on correct principles. In many cases a rectangular furnace, originally designed for a flat fuel bed radiating heat vertically may, with few material changes in construction, be readily converted to burn pulverized fuel. Such an installation may operate with good efficiency but it is not so well adapted to successful automatic control as a furnace especially designed for this class of fuel.

Means of Control

Steam requirements must necessarily control the quantity of fuel and air admitted to the furnace and the CO_2 per cent, that it is desired to maintain, must determine the ratio of fuel to air in the mixture.

Since steam pressure and steam flow bear a definite relation to each other we can actuate the control by either one or both of these agencies and the nature of the steam requirements will in most cases determine which it is preferable to use. The author of an article on combustion control in a previous issue makes the broad statement that "It must be definitely recognized that the primary impulse must not in any case be a device aimed at holding constant pressure at the boiler outlet or anywhere else."

He also compares regulation of a boiler to constant pressure to an isochronous steam engine or turbine governor.

It is of course recognized by power plant engineers that neither steam pressure nor steam flow can be maintained at all times absolutely in accordance with requirements. It is desirable, however, that steam pressure be held fairly close to the maximum in order to maintain maximum economy in the operation of the prime movers, as there is a direct relation between steam pressure and steam economy.

No exact line of comparison can be drawn between the governing of steam engine or turbine speed and the regulation of combustion. The impulses of the steam engine governor meet with almost instant response, while with combustion control an appreciable length of time elapses before the boiler responds to a change in the amount of fuel burned in the furnace. Therefore, in the design and operation of combustion control equipment, the fact should be considered that time is required for the fuel to become ignited and burned and for the heat thus liberated to enter the water and produce steam. If a variation of one pound in steam pressure actuates the regulating device to change the quantity of fuel being supplied to the furnace, a further change in pressure will occur before the different furnace conditions will have had time to bring the pressure back to normal again.

I doubt if primary impulse from steam flow instead of steam pressure would cause the response to be more rapid, or result in any improvement of the regulation.

Method of Operation

In order to obtain the best results in regulating combustion, the method of control should be such that a change in the boiler pressure will be arrested as quickly as possible and brought back to normal at a rate that will not cause it to pass beyond normal, otherwise a pulsating action will take place. This same action will occur if we attempt to obtain close regulation by operating the control through a narrow range of steam pressure or steam flow.

In order that a change in steam pressure may be quickly arrested, the automatic control should be so arranged that when the device acts to vary the amount of fuel and air mixture being supplied to the furnace, the change in quantity will be made as quickly as possible; and the amount of increase or decrease in the quantity will be in proportion to the rapidity of the initial change in steam pressure which actuated the control. In addition, the change in the quantity of fuel and air mixture should be slightly more or less than is necessary to hold the pressure at normal under the changed conditions; but immediately the drop or rise in pressure is arrested, the regulator should act gradually to taper off this quantity to an amount just sufficient to carry the normal pressure under the new conditions. The accompanying curves show the relation between steam pressure and fuel supply with this method of operating combustion control.

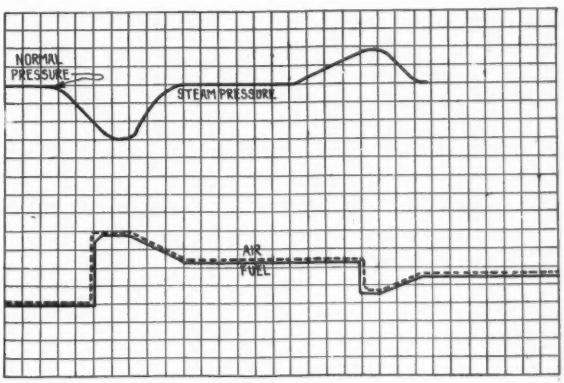
The initial impulse, whether it be steam pressure or steam flow, should control directly the fuel supply, air supply, and uptake draft. This may be called the primary control. The ratio of fuel to air in the mixture will, as a rule, remain fairly constant and will be determined by the per cent of CO_2 it is desired to maintain. In order to maintain efficient combustion, however, it may be desirable to vary this ratio when the amount of fuel being supplied to the furnace is rapidly increased or decreased. That is, it may be necessary to precede an increase in the supply of fuel mixture by a momentary increase in the air supply and to precede a decrease in fuel mixture by a momentary decrease in the amount of fuel.

Draft maintained at any given time depends upon the amount of fuel and air entering the furnace but as it is desirable that the furnace draft should approach as closely as possible to zero, the primary control will be influenced by the pressure within and without the furnace walls.

There will then be primary control of the fuel, air, and uptake draft, and secondary control of the fuel and air ratio and of the furnace draft.

The device with which automatic regulation is accomplished should be a simple and sturdy piece of apparatus, but accuracy and good workmanship should not be sacrificed otherwise sensitiveness and reliability can not be attained.

Automatic regulation can successfully control combustion but it is, of course, realized that no mechanical device can anticipate a possible demand on the boiler. The regulator can not act until the conditions have actually changed and the success of the control will then depend upon the manner in which the changed conditions are met.



CURVES INDICATING RAPID CHANGES IN AIR AND FUEL SUPPLY TO CARE FOR VARIATION OF STEAM PRESSURE FROM NORMAL

The fact that some existing installations may not lend themselves readily to automatic combustion control need be no criterion of what it is possible to accomplish with proper design of furnace and regulator.

Chicago Uses Superpower

On Dec. 11, 1924, the peak demand in Chicago was 688,000 kw. or 922,250 hp. and for the 24 hr. the energy used was 10,306,770 kw-hr. That such a demand was met without incident is proof of the wise foresight and planning of the Commonwealth Edison Co. during past years.

At present, almost $200,000,000 is invested in the system of the company which has 43,000 stockholders in the territory supplied, and furnishes current to 750,000 customers. The new Crawford Ave. plant, now partly in operation, will, when completed, have a capacity nearly as great as the total possible power from the much-discussed Muscle Shoals. Besides, there is interconnection of Chicago stations with central supply stations in Illinois, Indiana, Iowa, Kentucky, Michigan and Wisconsin by means of which power can be interchanged as may be desirable—a real superpower system created by normal, natural development to meet demand and emergencies and to secure the greatest economies possible.

Standing Torque of Small Turbines

STANDING AND RUNNING TORQUE APPROACH EACH OTHER AS A TURBINE BECOMES LESS EFFICIENT. BY J. Y. DAHLSTRAND*

CERTAIN APPLICATIONS are made of steam turbines where the standing or starting torque and its relation to power output of the machine become of special interest and importance. This is often the case when the turbine is used for direct mechanical drive.

Small turbines used for this purpose are usually one of two types; namely, impulse or a combination of velocity

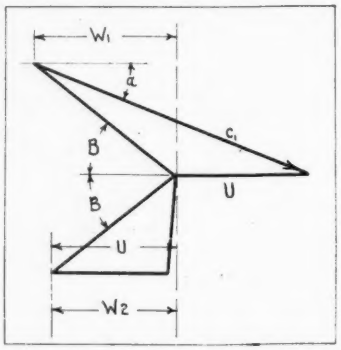


FIG. 1. VELOCITY DIAGRAM OF A SINGLE ROW TURBINE STAGE

and impulse, the latter being known as velocity and pressure stage machines. To begin this study let us first consider the velocity diagram, Fig. 1, of a typical impulse turbine. In order to arrive at the theoretical conditions which exist we must bear in mind that the following factors are important:

V = Total flow of steam, lb., per sec.
c = Theoretical steam velocity in each stage
c_1 = Actual steam velocity in each stage
U = Blade velocity in each stage
a = Nozzle angle
B = Bucket angle
x = Bucket coefficient
y = Nozzle coefficient
n = Number of stages

At the start certain assumptions are made to simplify the problem. These are, (1) Bucket inlet and outlet angles are assumed equal, (2) Steam entrance is assumed free from shock, (3) All stages are assumed to have the same wheel diameter and the same number of heat units taken out in each stage and (4) The turbine is of the pure impulse type having one row of rotating blades in each stage and without reversing chambers. After relations have been established under these conditions, they can readily be determined for others.

Again referring to Fig. 1, it will be noticed that the velocity component in the direction of the wheel tangent is equal to the sum of the components W_1 and W_2;

since $W_1 = c\ y \cos a - U$ ..... (1)

and $W_2 = X\ (c\ y \cos a - U)$ ..... (2)

then the sum of these two components would be

$Ew = (1 + X)\ (c\ y \cos a - U)$ ..... (3)

$P = V/g\ (1 + X)\ (c\ y \cos a - U)$ ..... (4)

and the total force in all the stages of the turbine would be

$P\ n = V/g\ (1 + X)\ (c\ y \cos a - U)$ ..... (5)

In the last two equations g is a constant and is equal to 32.2 ft. per sec. per sec., or the acceleration due to the force of gravity.

If the pitch diameter of the turbine wheel is D inches and the torque is M_t in. lb., the running torque M_t at a speed equal to U ft. per sec. will be

$$M_t = \frac{DVn}{64}\ (1 + X)\ (c\ y \cos a - U) \quad \ldots\ldots (6)$$

Now if we assume factors as follows; a = 14 deg., B = 27 deg., X = 0.85 and y = 0.95, then equation (6) becomes

$M_t = 0.0289\ DVn\ (0.92c - U)$ ..... (7)

EFFECT OF CHANGE IN BUCKET LOSSES

In the value for torque expressed above, there has been no account given to two factors namely, windage losses

TABLE I. TABULATED RESULTS OF TESTS AND CALCULATED FIGURES FOR A NINE-STAGE TURBINE

Test of nine stage turbine. D = 15.375 in. Total steam flow, 130 lb., Ring pressure = 4690 lb. per hr. V = 1.3 lb. per sec.						Calculated figures from equations	
Steam Ring Press. Lb. Ga.	Super Heat	Exh. Press.	R.P.M.	Brake horsepower from test	Running Torque from test	M_t from Equation No. 7	Standing Torque from Equa. No. 10
130	Dry & Saturated	Atm.	4200	164.8	2470	3125	
"	" "	"	3900	163	2638	3230	
"	" "	"	3600	160.5	2800	3335	
"	" "	"	3300	155	2950	3440	
"	" "	"	3000	148	3100	3545	
"	" "	"	2700	139	3240	3650	
"	" "	"	2400	129	3380	3755	
"	" "	"	2100	117.5	3520	3860	
"	" "	"	1800	104.5	3660	3966	
"	" "	"	1500	90.5	3795	4072	
"	" "	"	1200	75	3930	4178	
"	" "	"	900	58.2	4075	4284	
"	" "	"	600	40.	4210	4390	
Standing Torque						4600	4510

and changes in bucket losses for changed relation between U and c. The first of these factors, however, disappears for zero speed and the second diminishes only about 8 per cent according to test data confirmed by several writers for a change in relative bucket entrance angle of from 27 deg. to 14 deg., the actual bucket angle being 27 deg.

If in equation (6) we take U, the blade velocity as equal to zero and change the value of x to 0.77, in place of 0.85, then the standing torque will be expressed by the simple formula:

$M_{st} = 0.0255\ DVnc$ ..... (8)

or since $c = 223.7 \sqrt{\frac{WK}{n}}$ ..... (9)

where W is the total heat in B.t.u. available in adiabatic expansion for use in the turbine and K is the reheat factor;

$$M_{st} = 1275\ \frac{DVWK}{c} \quad \ldots\ldots (10)$$

*Mechanical Engineer, Standard Turbine Corp.

Thus, it is apparent that the standing torque for given steam conditions is increased for decreased steam velocity and increased with an increased number of stages.

It is interesting to note that the value of the running torque, found in equation (7), is a straight line function of U and the operating speed. To express the actual running torque, equation (7) would have to be corrected for the difference in bucket efficiency which correction would still, however, leave the relation a straight line function. It would also have to be corrected for windage friction which would bend the line slightly. The value of the windage losses depends on a variety of factors such as bucket lengths, disk surface, etc., which would render it difficult to lay down general rules for its value.

Shown in Table I is a series of test figures obtained with a turbine of the type under consideration. This table

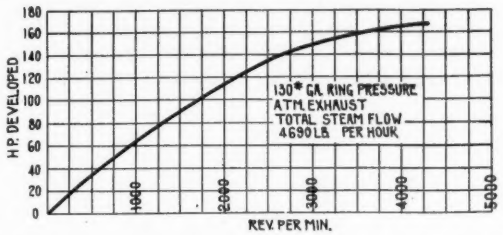


FIG. 2. STEAM TURBINE POWER-SPEED CHARACTERISTICS OBTAINED BY TESTS

shows the horsepower developed for given steam conditions and with varying speeds. The torque was figured from the horsepower developed by the following equation:

$$M_t = 63{,}025 \frac{\text{horsepower}}{\text{r.p.m.}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

The results obtained are shown in Fig. 2 and Fig. 3, in which are shown the figures for starting torque as obtained from equation (10) and the figures for running torque as obtained from equation (7). There are also shown dotted lines which represent the running torque as corrected for varying bucket efficiency and the reheat factor.

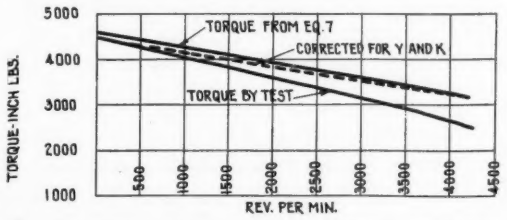


FIG. 3. STANDING AND RUNNING TORQUE OBTAINED BY TEST AND FROM EQUATIONS

Equation (10) will hold with sufficient accuracy for combination velocity and pressure stage machines with one velocity stage and five or more pressure elements. For machines having two rows of velocity stages, they must be considered as equivalent to 2.1 times the values of n used for a one-row pressure stage machine and c figured accordingly. For velocity stage turbines with two rows of revolving buckets in each stage, the value of M_t is approximately expressed by the equation:

$$M_t = 0.068\ D.V.n.\ (0.492\ c - U) \quad \ldots\ldots\ldots\ldots\ldots\ldots (12)$$

$$M_{st} = 0.0334\ D.V.n.c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (13)$$

these figures having been arrived at by testing a large number of machines. The equation (12) will be altered somewhat with variations of such factors as partial admission, under-expansion of nozzles, but the value given represents an average figure which will be true with a narrow percentage. Table II gives the test figures of a single-stage Curtis type turbine.

Check May Be Made by Test

It is interesting to note that the relation between the standing torque and running torque for a given speed having been once established in actual test may afterwards be used as a basis for check tests of the output by measuring the standing torque, since practically anything which can happen to a turbine, such as clogged nozzles or blades, eroded blades, leakage in the interior of a turbine, or the like, will have proportionate effects on standing and running torque. The standing torque of a turbine as compared with its running torque depends, as will be noted, mainly on the relations between blade and steam velocities.

Since in the single-row impulse turbines the maximum efficiency is attained at the point where the steam velocity is slightly more than twice the blade speed, it will be noted that a standing torque of somewhat less than twice the running torque at proper operating speed is about the maximum which can be obtained with an efficient turbine.

TABLE II. TEST FIGURES AND CALCULATED RESULTS OF A SINGLE-STAGE TURBINE WITH TWO REVOLVING ROWS OF BUCKETS

Test of single stage turbine with two revolving rows of buckets. D = 28 in. Total steam at 150 lb., Ring pressure = 5050 lb. V = 1.405 lb. per sec.						Calculated figures from equations	
Steam Ring Press. Lb. Ga.	Super Heat	Exhaust Press.	R.P.M.	Brake horse-power from test	Running Torque from test	U_t from Equation No. 12	Standing Torque from Equation No. 13
150	Dry & Saturated	Atm.	3600	149	2610	2660	
"	" "	"	3000	137	2880	2880	
"	" "	"	2500	121	3050	3035	
"	" "	"	2000	101	3180	3190	
"	" "	"	1800	93	3250	3250	
"	" "	"	1500	80	3360	3345	
"	" "	"	1350	73.5	3430	3397	
"	" "	"	1200	66	3460	3450	
"	" "	"	1000	55.5	3500	3500	
"	" "	"	800	46	3625	3590	
Standing Torque							3840

The more inefficient a turbine is the nearer lies its standing torque to its running torque, the lines of torque running closer to parallel with the base line, which also accounts for the fact that an inefficient turbine will become self-loading at a higher speed than an efficient one, other conditions being the same. This self-loading takes place when the effective running torque becomes zero.

Fuel oil storage tanks should be entirely closed excepting for the safety vent, as the evaporation of the oil is a larger factor than is generally supposed. This is especially true in cases where the tanks are exposed to the heat of the sun.

A spirit of friendly rivalry and honest competition between the shifts of a plant, for efficiency in quality and quantity of output is an excellent thing.

Test of a Prosser Type Steam Engine*

DESCRIPTION OF THE PROSSER RECIPROCATING ENGINE AND RESULTS OF 53 TESTS MADE OF IT AT PURDUE UNIVERSITY. BY L. V. LUDY

ONE OF THE greatest sources of loss in the operation of the ordinary reciprocating steam engine is the initial condensation of steam upon the cool walls of the cylinder and piston. In modern engines the alternate heating and cooling effect is much less than formerly, but the loss due to this cause is still great. In studying this phase of cylinder loss in steam engines, J. G. Prosser developed ideas which have been incorporated in an engine

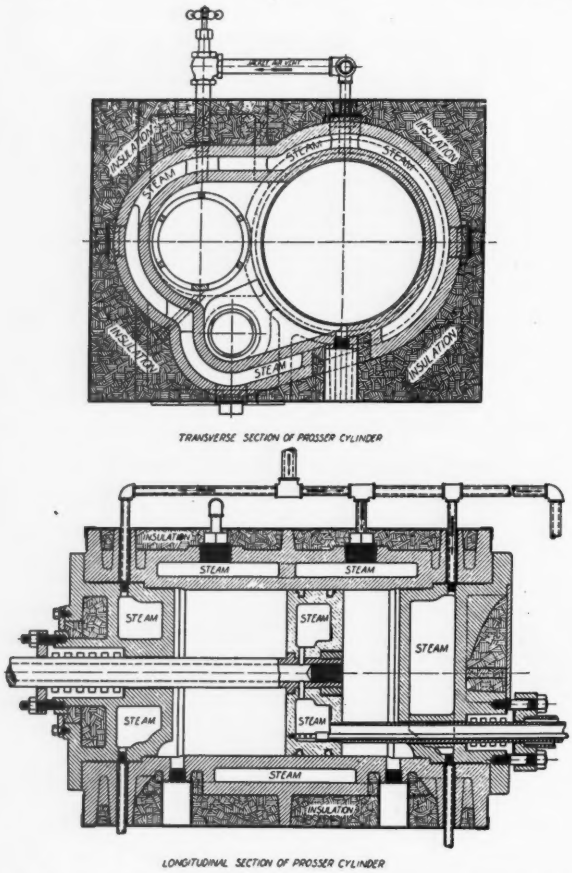


FIG. 1. SKETCH ILLUSTRATING ARRANGEMENT OF JACKETS

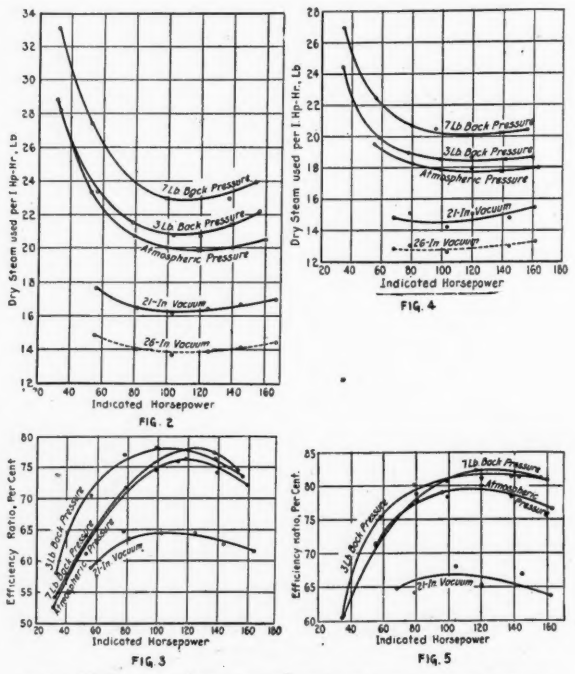
known as the Prosser Steam Engine. In order to determine the performance of this engine, a series of 53 tests were made of it at Purdue University. The results of these tests are contained in this article.

Most engineers are familiar in a general way with the thermodynamic losses which occur in the cylinder of a reciprocating steam engine. For this reason it seems unnecessary to refer in detail to them. It is perhaps sufficient to state that in the practical operation of a steam engine a certain amount of heat must be rejected in the exhaust, which under a given set of conditions cannot be reduced; and that the transfer of heat to and from the steam between the initial and final temperature conditions results in making unavailable a large amount of heat for transfer into useful work. This latter loss can be reduced. Theoretically the heat change in the cylinder, between the initial and final temperature conditions should occur adiabatically; that is, in the ideal expansion no heat is lost as heat. In practice the adiabatic change cannot be secured, but by applying certain thermodynamic principles it can be made to approach the ideal condition.

The methods commonly employed in approaching the above conditions are

a The use of superheated steam
b The use of steam jackets
c The use of compound cylinders.

The first two methods are made use of in the Prosser engine. Neither of these methods is new, but the steam-jacketing feature is carried out differently, in that in addition to the cylinder, valve case and heads being provided with steam jackets, the pistons and rod are hollow and supplied with high pressure steam for jacket purposes.



FIGS. 2-5. RESULTS OF TESTS SHOWN GRAPHICALLY

Fig. 2. Water Rate Curves, Saturated Steam. Fig. 3. Efficiency Ratios, Saturated Steam. Fig. 4. Water Rate Curves, Superheated Steam. Fig. 5. Efficiency Ratios, Superheated Steam.

THE PROSSER ENGINE

The general appearance of the Prosser engine is quite similar to engines built by several well-known manufacturers. That used in this investigation was a single-cylinder, simple, horizontal, high-speed engine designed for direct connection to an electric generator. Provision is made for supplying high-pressure steam to the steam jacket surrounding the cylinder and valves, to the hollow space of each cylinder head, and to the piston and piston rod. The general arrangement of all jackets is indicated in the sketch presented as Fig. 1. The manner in which steam is admitted to the hollow portion of the piston and piston rod, through the piston tail rod, is indicated in Fig.

*Abstract of a paper presented at the A. S. M. E. meeting in New York, Dec. 1, 1924.

TABLE I. AVERAGE OBSERVED AND COMPUTED RESULTS, SATURATED STEAM

	Exhaust condition											Exhaust condition											
	21 in. vacuum					Atmospheric exhaust						3 lb. back pressure						7 lb. back pressure					
1 Trial No.	37	36	25	27	26	29	40	41	42	43	44	45	46	47	48	49	50	59	58	51	52	53	54	55	56	57
2 Net brake load, lb.	250	300	400	500	600	700	300	400	500	600	650	700	300	400	500	600	650	700	700	300	400	500	600	650	700	
3 Duration of trial, min.	60	60	60	60	60	60	60	60	60	60	55	47	60	60	60	60	60	60	60	60	60	60	60	60	60	60
4 Steam-line pressure, lb. per sq. in. gage	132.15	133.92	133.62	134.15	136.92	133.23	135.38	130.62	127.46	124.15	121.77	126.69	138.54	136.15	135.38	136.69	136.38	132.92	134.0	132.62	132.69	133.92	132.00	134.38	134.00	130.46
5 Barometric pressure, in. hg.	29.38	29.14	29.09	29.14	29.38	29.27	29.35	29.46	29.44	29.45	29.46	29.28	29.31	29.27	29.28	29.39	29.4	29.15	29.34	29.42	29.35	29.39	29.60	29.17	29.14	
6 Revolutions per min.	217.95	217.5	214.5	213.0	210.2	208.5	215.5	214.2	210.2	208.5	208.5	207.4	217.4	214.9	211.6	210.35	208.3	206.5	204.9	214.9	212.4	210.6	209.3	207.1	205.0	203.55
7 Room temperature, deg. fahr.	87.9	86.6	89.2	87.8	89.5	92.5	94.2	91.8	92.4	91.9	92.0	94.7	87.0	89.9	90.3	92.9	93.8	89.4	97.8	79.8	83.7	86.7	97.4	95.5	94.5	97.4
8 Temp. of steam at throttle, deg. fahr.	471.2	466.8	463.0	454.7	457.2	457.2	464.3	464.1	457.2	454.0	471.5	461.5	469.5	460.4	466.5	464.0	454.5	460.0	465.8	460.7	466.5	459.0	462.2	454.7	456.6	454.4
9 Temp. of steam in jackets, deg. fahr.	351.0	350.0	349.2	350.8	351.2	350.2	350.4	351.8	349.5	348.9	353.8	356.3	357.6	357.6	356.4	356.5	355.5	355.6	355.0	356.2	356.2	357.0	356.8	355.8	355.5	355.5
10 Temp. of steam in exhaust, head end, deg. fahr.	153.8	153.4	152.9	150.0	150.7	157.7	218.5	214.9	213.0	214.7	214.8	215.0	278.5	275.4	274.5	274.7	274.0	274.1	274.0	280.1	277.7	274.8	274.2	274.1	274.0	274.1
11 Temp. of steam in exhaust, crank end, deg. fahr.	153.0	152.4	152.2	150.2	150.3	156.9	213.2	212.3	212.9	212.5	213.3	213.8	246.4	225.0	221.0	221.0	220.2	221.2	221.2	254.4	240.2	231.8	231.7	231.7	231.7	231.7
12 Pressure in exhaust, lb. per sq. in. gage							0	0	0	0	0	0	3	3	3	3	3	3	3	7	7	7	7	7	7	7
13 Vacuum in condenser, in. hg.	29.98	21.12	21.08	29.91	29.92	21.07																				
14 Steam used per hour, from condenser, lb.	967.0	1133.0	1437.5	1746.0	2087.0	2430.5	1626.5	1395.5	1711.5	2085.0	2260.5	2660.5	792.5	1100.0	1468.0	1793.5	2163.0	2553.5	2940.5	908.75	1202.5	1586.0	1910.0	2286.5	2717.5	3107.5
15 Steam used per hour, from jackets, lb.	54.5	61.5	69.0	64.0	58.5	65.0	56.5	57.0	69.5	68.5	71.5		46.0	56.0	61.5	63.0	62.5	59.5	62.0	43.5	51.0	63.0	58.0	60.5	72.5	69.0
16 Steam line pressure, lb. per sq. in. abs.	146.28	148.22	147.86	148.46	151.35	147.96	149.91	145.00	141.92	138.62	136.24	151.07	152.94	152.68	149.91	151.13	150.62	148.25	148.30	147.19	146.34	148.14	150.54	148.69	148.33	147.77
17 Degrees superheat at throttle, deg. fahr.	114.5	109.2	105.62	97.02	100.4	99.9	106.0	108.2	113.0	111.7	115.0	102.4	109.5	100.6	102.1	104.0	95.6	111.4	106.2	112.7	108.7	101.7	109.6	96.9	99.3	98.1
18 Total dry steam used per hour, lb.	1021.5	1194.5	1477.5	1810.0	2145.5	2504.5	1683.0	1452.5	1780.0	2153.5	2332.0	2660.5	838.5	1156.0	1469.5	1856.5	2225.5	2613.0	2975.5	952.25	1253.5	1649.0	1968.5	2346.0	2790.0	3176.5
19 Mean effective pressure, head end, lb. per sq. in.	24.84	30.31	39.92	45.77	55.85	67.68	22.51	32.88	40.08	48.01	55.23	65.27	15.6	24.04	31.70	40.07	47.82	56.55	63.58	16.59	24.82	33.77	39.96	47.94	56.26	62.72
20 Mean effective pressure, crank end, lb. per sq. in.	29.78	21.77	29.27	34.83	47.64	48.71	14.40	20.87	27.26	34.23	39.97	47.41	6.96	14.77	20.95	27.35	34.07	41.99	48.99	6.99	15.84	20.59	25.74	33.81	39.86	47.36
21 Indicated horsepower, head end.	38.12	46.47	60.30	69.61	82.67	92.05	34.16	49.50	59.23	70.40	81.09	98.47	23.86	36.38	47.24	59.53	69.62	82.23	91.74	25.11	37.29	50.02	59.80	69.89	81.57	89.78
22 Indicated horsepower, crank end.	30.49	32.67	43.47	51.34	61.84	70.08	21.41	30.84	39.68	49.24	57.49	67.85	10.44	21.90	30.59	39.64	49.83	59.70	67.99	10.23	18.80	29.92	37.37	48.31	56.63	66.42
23 Total indicated horsepower.	68.61	79.09	103.77	120.99	144.51	162.11	55.57	80.44	98.91	119.73	138.58	166.32	34.32	58.28	77.83	99.01	119.65	141.93	159.73	35.34	56.09	79.94	96.97	117.20	138.20	156.15
24 Dry steam used per i.hp.-hr.:																										
By cylinder, lb.	14.92	14.92	13.85	14.55	14.45	15.05	18.47	17.31	17.79	17.42	17.29	17.51	23.00	18.87	18.89	17.81	18.11	17.99	18.22	25.71	21.54	19.84	19.80	19.50	19.06	19.90
By jackets, lb.	0.80	0.78	0.69	0.53	0.40	0.43	1.02	0.71	0.61	0.50	0.45	0.44	1.34	1.00	0.79	0.64	0.52	0.42	0.41	1.23	0.91	0.81	0.61	0.52	0.53	0.44
Total, lb.	14.92	15.10	14.24	15.08	14.85	15.45	19.49	18.02	17.90	17.92	17.74	17.95	24.43	19.87	19.68	18.45	19.61	18.41	18.63	26.94	22.45	20.65	20.40	20.02	20.19	20.34
25 Efficiency ratio, per cent.	64.9	64.1	66.0	65.2	66.9	69.7	71.2	73.0	73.3	69.0	78.4	76.0	69.4	75.2	80.2	80.7	81.1	81.3	80.2	69.6	72.7	78.3	79.6	82.5	81.3	81.1

2. The condensate which accumulates in the cylinder and valve-case jacket and the cylinder-head jackets is conducted by gravity through suitably arranged pipes to a trap below. The condensate from the hollow piston rod and piston collects in the lower portion of the hollow piston and is removed through a small ball check valve and drain tube by the pumping action of the parts. Small hand-operated air cocks located in the outer portion of the piston rod and at other convenient places provide a means of removing air from the steam jackets. All surfaces of the cylinder block and heads exposed to the outside atmosphere are lagged with asbestos insulation in order to reduce radiation losses to a minimum.

The valve mechanism consists of two piston valves, one being 7 in. and the other 4 in. in diameter. The larger of the two, the main valve, is operated by an eccentric under control of the governor and regulates the admission, cutoff, and release of the steam. The smaller valve is operated by a fixed eccentric attached to the engine shaft, its function being to control the point of compression independently of the main valve.

The engine is equipped with a shaft governor of the Armstrong type, which regulates the stroke of the valve to suit load conditions by swinging the eccentric across the shaft. The governor is provided with a dashpot to dampen the vibration tendencies of the governor parts. Lubrication of the piston and valves is accomplished by means of a mechanical lubricator, while the crosshead, guides, valve rod sliding-block guides, eccentric, main bearing, crank and crosshead pins, etc. are lubricated by gravity through the sight-feed oilers. The outboard bearing of the engine shaft is lubricated by ring oilers.

Arrangement of Equipment for Tests

In anticipation of the test a concrete foundation was constructed, having a floor area of 12 by 15 ft. and a depth of 8 ft. Condenser equipment, an independently fired superheater, an absorption brake of the Prony type and all necessary gages, thermometers, condensate-weighing tanks, condensate-weighing scales, etc., were provided. The engine shaft, instead of carrying the armature of an electric generator, for which it was designed, was fitted with a heavy cast-iron brake pulley.

Steam was supplied through a 4-in. branch pipe from the steam main, the exhaust being conducted to a surface condenser through a 7-in. pipe. The condenser is fitted with a circulating-pump cylinder and a wet-vacuum-pump cylinder, both of which are direct-driven by a steam cylinder located between them. Circulating water was supplied from the service lines of the city water company through a 4-in. pipe. Located in the exhaust pipe above the condenser was an angle valve by the use of which any desired back pressure could be maintained. Two 1-in. globe valves connected by short nipples to the top portion of the condenser shell were opened for operation at atmospheric exhaust, and at times were opened slightly to assist in maintaining a constant vacuum.

For saturated-steam tests the steam was taken directly from the main steam line through the 4-in. branch pipe and separator mounted directly above the throttle. For superheated steam tests valves were adjusted in the main steam line so as to pass the steam through the superheater.

The condensate from the condenser was weighed in steel tanks mounted upon platform scales. That from the different jackets was conducted from manually operated traps through small pipes to a steel tank, partially filled with cool water, mounted upon platform scales where it could be weighed, the purpose of the cool water being to prevent vapor loss.

A throttling calorimeter was connected to the steam supply pipe immediately above the throttle, a standard sampling tube being used. New Crosby steam-engine indicators were used in determining the indicated horsepower developed. A counter was attached to the free end

TABLE II. AVERAGE OBSERVED AND COMPUTED RESULTS, SUPERHEATED STEAM

	Exhaust condition												Exhaust condition													
	21 in. vacuum						Atmospheric exhaust						3 lb. back pressure						7 lb. back pressure							
1 Trial No.	7	8	24	9	10	11	28	29	30	31	32	25	34	15	17	16	18	19	20	22	14	33	13	12	96	21	23
2 Net brake load, lb.	200	300	400	500	600	700	300	400	500	600	650	700		300	300	400	500	600	700	700	300	400	500	600	650	700	
3 Duration of trial, min.	60	60	60	60	60	45	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
4 Steam-line pressure, lb. per sq. in. gage	153.62	121.9	134.15	130.8	133.3	133.6	133.46	133.77	136.69	137.1	133.31	136.46	140.0	131.62	130.15	128.08	135.0	136.0	133.23	134.52	130.92	136.38	135.46	134.16	134.92	133.69	133.54
5 Barometric pressure, in. hg.	29.95	29.42	29.86	29.42	29.37	29.27	29.29	29.29	29.28	29.26	29.24	29.21	29.10	29.27	29.14	29.13	29.6	29.6	29.31	29.04	29.27	29.22	29.23	29.25	29.21	29.23	29.25
6 Revolutions per min.	219.1	218.4	214.2	213.5	211.6	209.7	219.0	214.85	214.7	211.6	209.9	209.7	209.7	217.6	214.6	211.7	210.7	208.4	207.2	205.1	218.1	215.2	213.0	210.8	209.6	209.4	208.2
7 Calorimeter pressure, lb. per sq. in. gage	4.96	5	5	4.98	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
8 Calorimeter temperature, deg. fahr.	274.8	270.3	272.5	269.0	271.5	273.4	277.8	270.3	278.0	279.5	279.3	279.4	278.5	271.6	272.1	272.6	272.2	272.9	273.0	274.7	271.6	278.3	279.6	277.6	279.0	277.0	280.9
9 Room temperature, deg. fahr.	90.5	87.5	89.5	87.5	85.0	89.1	85.7	85.5	79.2	81.9	83.8	90.3	92.8	88.5	84.0	87.9	83.7	83.7	82.2	85.0	86.4	92.8	90.2	88.5	91.5	82.6	86.8
10 Temp. of steam at throttle, deg. fahr.	357.5	357.7	357.8	356.7	357.7	356.1	355.8	356.9	357.5	356.9	354.8	356.5	357.4	357.7	356.3	356.2	356.9	356.2	357.8	357.6	357.1	357.9	356.9	356.0	356.5	356.6	357.9
11 Temp. of steam in jackets, deg. fahr.	354.0	353.4	354.0	353.6	356.1	356.5	354.6	354.6	356.1	356.0	356.0	356.0	356.0	354.2	353.2	353.6	353.6	354.6	354.6	354.6	353.6	353.3	353.6	353.6	353.6	353.4	353.4
12 Temp. of steam in exhaust, head end, deg. fahr.	152.7	155.5	152.5	156.7	156.1	156.6	213.8	213.2	214.5	215.0	215.0	215.0	215.0	227.3	223.0	223.1	224.0	224.0	223.9	223.8	227.6	224.2	224.1	223.5	223.2	223.3	223.0
13 Temp. of steam in exhaust, crank end, deg. fahr.	151.6	154.9	151.5	155.7	155.0	155.0	215.0	215.0	215.0	211.0	212.0	212.0	213.0	224.0	220.0	219.0	220.0	222.0	220.0	221.0	224.0	220.0	220.0	220.0	220.0	220.0	220.0
14 Pressure in exhaust, lb. per sq. in. gage							0	0	0	0	0	0	0	3	3	3	3	3	3	3	7.05	7	7.08	7	7	7	7
15 Vacuum in condenser, in. hg.	21.35	21.1	21.01	21.06	21.02	20.91																					
16 Steam used per hour from condenser, lb.	911.5	1267.5	1618.5	1995.0	2332.75	2716.0	800.0	1146.0	1550.5	1891.0	2282.0	2716.0	3190.5	872.5	1228.0	1603.5	1986.0	2404.0	2867.0	3139.0	980.0	1342.5	2002.25	2155.75	2538.0	3044.5	3660.0
17 Steam used per hour from jackets, lb.	95.0	109.0	116.5	129.0	124.5	152.5	90.0	105.5	126.0	124.0	138.0	137.5	143.0	94.5	117.0	129.0	131.0	137.5	145.5	163.0	98.5	129.0	135.0	132.75	140.0	153.0	167.0
18 Steam-line pressure, lb. per sq. in. abs.	167.86	136.36	148.38	145.25	147.73	147.83	147.80	148.11	151.07	151.47	147.67	150.81	154.34	146.9	144.66	142.99	149.54	150.54	147.98	149.22	145.20	150.73	149.93	148.62	148.98	148.05	147.90
19 Quality of steam at throttle, per cent.	98.21	98.07	98.12	97.99	98.27	98.32	98.40	98.46	98.39	98.65	98.44	98.34	98.09	98.19	98.2	98.12	98.15	98.32	98.22	98.08	98.44	98.46	98.36	98.44	98.37	98.40	
20 Total dry steam used per hour, lb.	986.50	1349.25	1683.13	2048.97	2414.74	2800.4	800.4	1232.48	1640.51	1984.37	2370.6	2810.05	3228.16	948.53	1329.66	1692.48	2086.97	2494.46	2967.37	3450.31	1026.42	1448.44	2104.76	2252.41	2684.35	3145.36	3672.1
21 Mean effective pressure, head end, lb. per sq. in.	19.79	29.53	38.25	45.76	55.39	63.56	14.62	21.08	27.99	29.39	47.41	54.46	62.71	15.3	22.62	31.65	41.07	49.60	54.96	62.45	15.40	23.45	29.07	40.52	46.96	54.74	61.92
22 Mean effective pressure, crank end, lb. per sq. in.	17.13	23.96	30.56	37.04	42.68	48.24	5.76	13.5	20.86	27.72	34.13	41.69	48.45	6.77	15.17	21.41	27.28	33.90	40.55	46.46	6.36	12.98	20.23	26.84	32.73	40.71	46.38
23 Indicated horsepower, head end.	30.39	45.42	57.72	70.29	82.25	93.59	22.55	32.80	43.56	56.09	69.75	79.63	90.84	23.44	34.38	47.19	60.79	71.52	80.08	90.59	23.37	34.97	47.61	60.06	69.90	79.19	89.51
24 Indicated horsepower, crank end.	26.99	35.85	45.19	54.57	62.51	71.19	8.70	20.01	30.90	40.47	49.21	59.72	69.72	10.17	22.46	31.27	39.57	49.81	57.89	66.78	9.16	17.92	29.62	39.34	46.68	57.70	64.72
25 Total indicated horsepower.	56.29	81.27	102.91	124.86	144.76	164.7	31.25	52.81	79.42	96.16	118.96	139.36	160.61	33.61	56.64	78.46	100.36	120.12	137.97	156.98	32.52	52.85	77.05	99.30	115.58	136.88	154.23
26 Dry steam used per i.hp.-hr.:																											
By cylinder, lb.	15.91	15.27	15.05	15.43	15.85	16.22	25.82	21.47	19.21	18.73	18.89	19.39	19.66	25.46	21.29	20.07	19.53	19.04	20.49	21.15	30.13	25.01	25.50	21.87	21.97	21.36	22.89
By cylinder, lb.	1.62	1.31	1.11	0.98	0.85	0.79	2.99	1.97	1.55	1.23	1.05	0.97	0.86	2.76	2.02	1.62	1.35	1.14	1.02	1.09	2.97	2.40	1.73	1.36	1.31	1.16	1.07
By jackets, lb.	17.53	16.58	16.16	16.41	16.70	17.01	28.81	23.54	20.77	20.01	19.95	20.17	20.54	28.22	23.32	21.67	20.81	20.76	21.43	22.18	33.10	27.41	27.23	23.21	23.28	22.56	23.96
Total, lb.	88.9	63.4	64.3	64.2	62.7	61.8	52.5	61.1	71.4	74.4	76.2	74.0	72.0	69.0	70.4	76.9	77.9	77.6	76.3	73.8	54.1	64.2	64.9	77.7	75.9	77.4	76.3
27 Efficiency ratio, per cent.																											

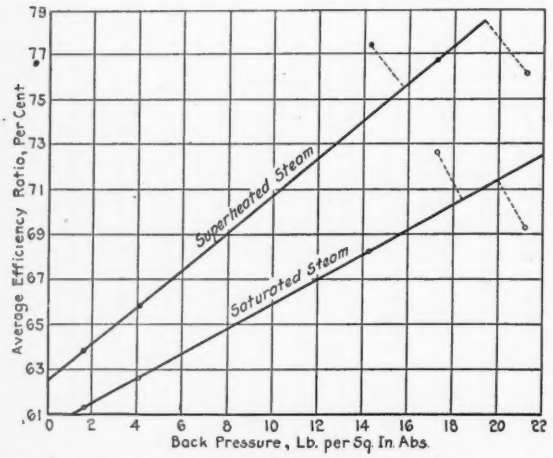


FIG. 6. RELATION BETWEEN EFFICIENCY RATIO AND BACK PRESSURE

of the engine shaft for determining the speed of the engine. A cast-iron exhaust manifold conducted the exhaust steam from both ends of the cylinder to the exhaust pipe. Thermometer wells were placed in each branch of the exhaust manifold to permit a temperature determination of the exhaust steam. A pipe attached at the midpoint of the exhaust manifold connected to a mercury vacuum gage for use when operating at a vacuum, and to a compound gage which indicated the back pressure in pounds per square inch or vacuum in inches. A thermometer well located on top of the cylinder casting, extending into the jacket space, was used in determining the temperature of the jacket steam. The temperatures of the steam entering the cylinder were read by a thermometer placed in a well-located immediately above the throttle. The steam pressure used was indicated by a pressure gage attached at a point just above the throttle.

The tests were conducted under the immediate direction of the author in the mechanical laboratory at Purdue University and were made possible through the cooperation of the Chandler & Taylor Co. of Indianapolis, Ind. Acknowledgment is made also of the able assistance rendered by R. G. Paddock, Instructor in Mechanical Engineering, and eight Senior Mechanical Engineering students.

Results

The average observed and computed results of the different trials as described above are presented in Tables I and II. The more important items of these tables are presented graphically in Figs. 2—5. In plotting these curves, the indicated horsepower was used as a common abscissa.

As the trials progressed, it was found that the condenser circulating-water supply was insufficient to maintain a vacuum of 26 in. of mercury under all load conditions. For this reason it became necessary to maintain as nearly as possible a vacuum of 21 in. for all load conditions to facilitate comparisons. Time did not permit of other trials at a vacuum of less than 21 in. from which it might be possible to estimate the steam consumption at a vacuum of 26 in., the following method was used.

The average efficiency ratio based upon the indicated horsepower was computed for each exhaust condition; the results secured are presented in Table III. In this table the first four items are those found from test conditions. These four items are plotted in Fig. 6 in order to determine the relation between the efficiency ratio and exhaust pressure. After finding the direction of the line it was extended to include the 26-in. vacuum, the results secured being given in Table III as item No. 5. Having found the estimated average efficiency ratio for a vacuum of 26 in., the following equation was used in correcting the steam consumption from a vacuum of 21 in. to a vacuum of 26 in.:

$$\text{Pounds dry steam consumed per i.hp-hr. at 26-in. vacuum} = \frac{(H_1 - H_2)\ \text{(at 21 in.vac.)}}{(H_1 - H_2)\ \text{(at 26 in. vac.)}} \times \frac{\text{ave. eff. ratio for 21 in.vac.}}{\text{ave. eff. ratio for 26 in. vac.}} \times \text{(Lb. dry steam consumed i.hp-hr. at 21 in. vac.)}$$

where H_1 = total heat contents per pound of steam at the initial condition

H_2 = total heat contents per pound of steam at the final condition, assuming adiabatic expansion.

The application of the above equation gave the results presented in Table IV, which are shown as a dotted line on Figs. 2 and 4.

Sample indicator cards are presented as Figs. 7 and 8.

TABLE III. AVERAGE EFFICIENCY RATIOS FOR DIFFERENT PRESSURES

Item	Exhaust pressure	Efficiency ratio: Saturated steam	Efficiency ratio: Superheated steam
1	21 in. vacuum = 4.00 lb. per sq. in. abs.	0.626	0.658
2	Atmosphere = 14.38 lb. per sq in. abs.	0.682	0.774
3	3 lb. gage = 17.38 lb. per sq. in. abs.	0.726	0.767
4	7 lb. gage = 21.38 lb. per sq. in. abs.	0.693	0.761
5	26 in. vacuum (estimated) = 1.62 lb. per sq. in. abs.	0.613	0.638

TABLE IV. ESTIMATED DRY STEAM USED PER INDICATED HORSEPOWER AT 26 IN. VACUUM—POUNDS

Saturated steam: Net brake load, lb.	Saturated steam: Dry steam per i.hp-hr. 21-in. vac.	Saturated steam: Dry steam per i.hp-hr. 26-in. vac.	Superheated steam: Net brake load, lb.	Superheated steam: Dry steam per i.hp-hr. 21-in.vac.	Superheated steam: Dry steam per i.hp-hr. 26-in. vac.
200	17.53	14.87	250	14.82	12.77
300	16.58	14.03	300	15.10	13.01
400	16.16	13.68	400	14.24	12.61
500	16.41	13.90	500	14.85	13.10
600	16.70	14.12	600	15.08	12.97
700	17.01	14.40	700	15.45	13.28

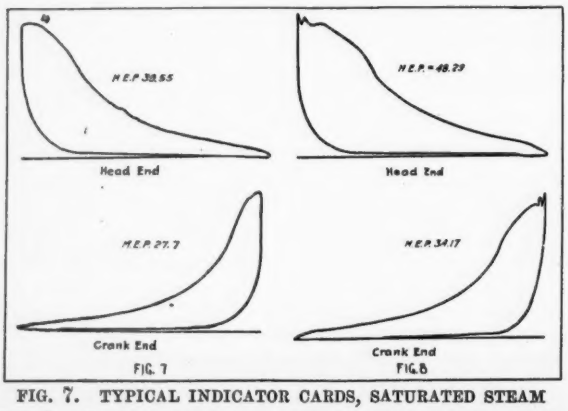


FIG. 7. TYPICAL INDICATOR CARDS, SATURATED STEAM

FIG. 8. TYPICAL INDICATOR CARDS, SUPERHEATED STEAM

Practical Applications of Hydraulic Principles---I

Explanation of Two Fundamental Laws, from Which Many Other Equations Can Be Deduced. Friction Losses Also Considered. By R. T. Livingston

UNTIL a comparatively short time ago, the average small plant operator scorned all forms of book learning. To him a technical school graduate was a man who was too lazy to work to attain the knowledge which he himself had secured by long hours of hard and hot work. With the increase of the size of plants and the added responsibility, however, the operator found himself faced with the necessity of getting some of this scorned book learning or getting a new job; in consequence, the operator today welcomes the opportunity of adding to his store of knowledge or of having some half-understood law explained to him in a new manner which clears up his difficulties.

To the initiated there are a large number of rules, equations and laws in hydraulics that are of daily use. It is interesting to realize that nearly all these rules are obtained from two elementary laws and if these two elementary laws are thoroughly understood almost any of these other equations can be deduced with ease. These two fundamental laws are the "Theorem of Continuity of Flow" and Bernouilli's Theorem.

This first law states that "in a closed system, with steady flow, the area of the pipe multiplied by the velocity in the pipe is always a constant and this constant is equal to the quantity flowing." To express this in mathematical symbols we say:

(1) $Q = A_1 \times V_1 = A_2 \times V_2 = A_3 \times V_3$ and so on. In other words, if a closed system is made up of a number of sections of different sizes, then if the velocity is high in one section it will be either higher or lower in the next section, depending upon whether the next section is smaller or greater than the previous one.

Bernouilli's Theorem states that "in a closed system, with steady flow, not considering friction the total energy —total head—is a constant and is equal to the sum of the three heads—static, velocity and elevational" and that these heads are mutually interconvertible. When written in the form of an equation, this becomes;

(2) $H = p_1 + (V_1^2 \div 2g) + z_1 = p_2 + (V_2^2 \div 2g) + z_2$

If we take friction into account, then the energy at a second point will be less than the energy at a first point by an amount equal to the amount of head lost in friction between these two points. With the use of these two laws taken together, almost any problem which will ordinarily be met in power plant work can be solved and the man who has a truly thorough understanding of them has an invaluable tool with which to work.

Solution of a Typical Problem

Let us now consider these two laws separately and see if we can make them clear. As probably the easiest way to understand any proposition is to do a problem, we will consider an arrangement of pipe shown in Fig. 1. We will say that the flow is occurring from left to right and that all the dimensions are actual distances and not a standard or nominal pipe size. It is assumed also that the velocity in the 3-in. section is 10 ft. per sec. The quantity flowing will obviously be:

$Q = A_1 \times V_1 = (3.14 \times 9 \div 4) \times 10 = 0.49$ cu. ft. per sec.

Then by the first law the velocities in the different sections can be obtained by dividing this amount—0.49—by the area of the pipe at the section considered; hence the velocities will be to each other inversely as the areas of those sections, or:

Velocity in 8-in. section = 1.4 ft. per sec.
Velocity in 6-in. section = 2.5 ft. per sec.
Velocity in 4-in. section = 5.63 ft. per sec.
Velocity in 1-in. section = 90.0 ft. per sec.

Obviously, if we have the velocity and the area we can compute the quantity or if we have a quantity that we wish to pass and do not wish to exceed a given velocity we can compute an area necessary. This law is fundamental and so obvious that it needs no further elaboration.

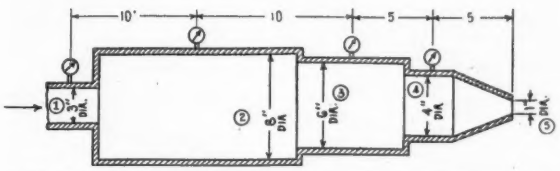


FIG. 1. ARRANGEMENT OF PIPE USED IN SOLUTION OF TYPICAL PROBLEM

Now let us consider Bernouilli's Theorem as applied to the same set-up. The pipe is horizontal, therefore the elevational heads will not vary, and we shall consider that there is no loss by friction. As a starting point we will take the edge of the nozzle, as here the only static pressure is that due to the atmosphere and we can disregard that as well as the elevational head. Therefore the total head must be equal to the velocity head, or, writing out the equation;

$H = p_5 + (V_5^2 \div 2g) + z_5 = 0 + (90^2 \div 64.4) + 0 = 125.6$ ft.

Then the static head at any point must be this total head minus the velocity head at that point, since all elevational heads are the same. Taking values of $V^2 \div 2g$ from the accompanying curve (Fig. 2) we have for static pressures:

At 3-in. section = 125.6 — 1.55 = 124.05 ft.
At 4-in. section = 125.6 — 0.49 = 125.11 ft.
At 6-in. section = 125.5 — 0.097 = 125.5 ft.
At 8-in. section = 125.6 — 0.03 = 125.57 ft.

If this pipe set-up were arranged vertically instead of horizontally we should have an entirely different story, as the effect of the difference of elevation would have to be taken into account. Let us compute the readings of the different static pressure gages if the nozzle still discharged the same amount of water. It will first be necessary to assume a datum plane, or reference plane and in order to avoid negative values of elevational head let us assume the plane through the static pressure gage in the 3-in. section. As before we will start at the nozzle:

$H = p_5 + (V_5^2 \div 2g) + z_5 = 0 + 125.6 + 30 = 155.6$ ft.

Now the static pressure at the various sections will be, as before, the total pressure minus the velocity heads but this time also minus the elevational heads. Thus:

4-in. section $p_4 = 155.6 - 0.49 - 25$ hence $p_4 = 130.1$
6-in. section $p_3 = 155.6 - 0.10 - 20$ hence $p_3 = 135.5$
8-in. section $p_2 = 155.6 - 0.03 - 10$ hence $p_2 = 145.57$
3-in. section $p_1 = 155.6 - 1.55 - 0$ hence $p_1 = 154.05$

Even if there were no flow there would still be a difference of reading on the various gages which would be equal to

the difference of elevation. If the pipe is on an angle instead of either vertical or horizontal, then the elevational differences will of course be the actual perpendicular distances which will be the distances along the pipe times the sine of the angle which the pipe makes with the horizontal.

Friction Losses Must Be Considered

In all these cases we have considered that there are no losses of any kind; however, this is not true and we must consider the effect of these losses. Just as whenever any

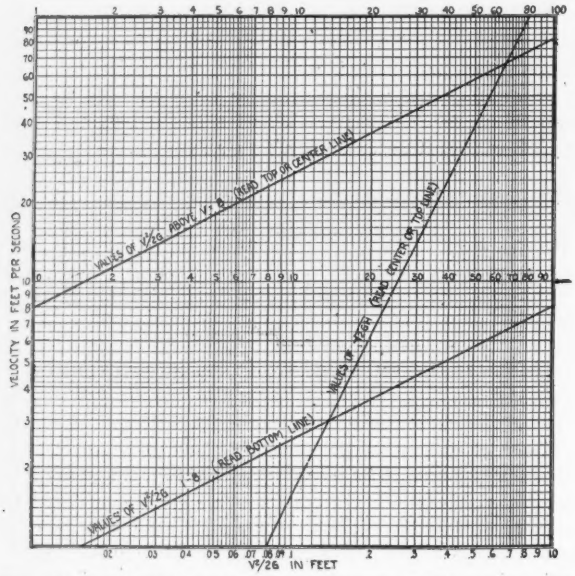

VELOCITY IN FEET PER SECOND
VALUES OF V²/2g ABOVE V² IN FEET (READ TOP OR CENTER LINE)
VALUES OF V²/2g IN FEET (READ BOTTOM LINE)
V²/2g IN FEET

FIG. 2. SHOWING RELATIONS BETWEEN VELOCITIES AND VELOCITY HEADS

part of a machine moves there is friction, so whenever water moves there is friction. Also whenever a moving body suffers a change of direction it takes a certain force to make this change; in a similar manner whenever water

LENGTHS OF PIPE TO GIVE SAME FRICTION LOSS AS CORRESPONDING FITTING

Fitting	Equivalent ft. of pipe
90 deg. ell	25-35
45 deg. ell	20-25
LR ells	15-25
Tees (out branch)	35-50

(These values are only approximations and the lower values given are for new pipe, the higher values for old pipe. The loss in LR ells depends upon the radius of the bend; the larger the radius the less the loss.)

suffers a change of direction, a force is necessary to make this change and this force is exerted at the expense of the energy—or head—of the water. It is hardly within the scope of this article to discuss these losses at any length other than to say that they depend, very nearly, upon the square of the velocity. We usually think of it as a loss of head represented by;

$$(3) \qquad h_f = K \times (V^2 \div 2g)$$

where K is a constant depending upon the diameter of the pipe, the character of the interior surface of the pipe and the particular type of loss considered.

Two of the most important types of loss that will be encountered in power plant work are the loss due to friction and that due to elbows, tees and other fittings. Figure 3 gives a chart with which the loss by friction may be obtained and in the table the losses due to ells and tees are given in equivalent lengths of straight pipe. The values given are sufficiently accurate for estimating and if any further information is desired the reader is referred to any standard text on hydraulics. In the particular set-up shown here, friction would have little effect, because of the short distances, but in any long or complicated set-ups it must always be taken into account.*

With a complete understanding of the above fundamental equations, it is unnecessary to remember any of the formulas for the Venturi meter or for nozzles. These

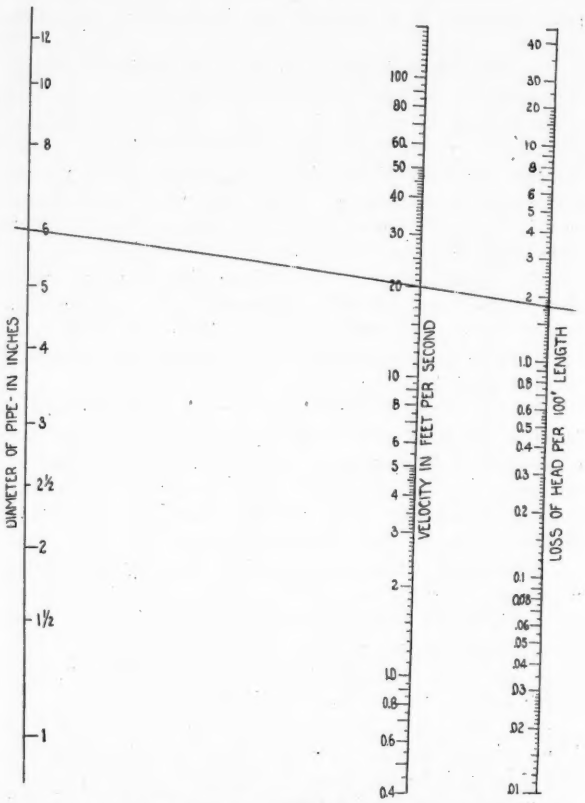


FIG. 3. ALINEMENT CHART, SHOWING RELATION BETWEEN PIPE SIZE, VELOCITY OF FLOW AND LOSS OF HEAD DUE TO FRICTION

formulas can be derived from the three given here, with the addition of factors termed co-efficients. In the next article of this series, the formula of the Venturi meter will be deduced and it is urged that any who are interested try to derive it themselves and compare it with the derivation given.

*The square of the velocity is really too high a power but it is quite generally used and is sufficiently accurate for our purpose. A full discussion of this phase of the question will be found in any book of advanced hydraulics such as Gibson or Lea.

DON'T EXPECT boiler safety valves to operate in emergencies when they are most needed if you neglect them at all other times.

Diesel Engine Applications Increase in America

LOCOMOTIVE, MARINE AND LARGE STATIONARY PLANTS PROVE PRACTICABLE. USE OF FUEL OIL DISTILLED FROM COAL MAY LEAD TO GREATER ECONOMY. BY R. C. DEMARY

AT THE present time, the steam driven unit is slowly but surely giving way to the heavy-oil engine. The latter can no longer be considered as in the development stage but must be looked on as an established source of power, thoroughly reliable and simple in construction. Its use has reached into every field, including the operation of railroads and ships.

Fuel economy, ability to start quickly and take full load at once, elimination of standby losses when shut down, absence of auxiliary equipment such as heaters, pumps, stokers, steam piping—all these give the oil engine an advantage. Furthermore, the thermal efficiency of many steam plants is rarely over 12 per cent, while that of the Diesel plant seldom falls below 32 per cent.

One general idea that seems to prevail is that the Diesel engine is made in small units only. The sizes manufactured range from 300 to 6000 b.hp. The illustration shows a four cylinder engine of American make. Each working cylinder has a 28-in. bore and 44-in. stroke and develops 550 b.hp. 28 in. by 44 in. This is a cylinder output never before attained in any other American Diesel engine and is responsible for the existence of the largest Diesel power plant on the American continent. This plant operated by the Moctezuma Copper Co., is located in Sonora, a state of Mexico and contains four 1250 b.hp. units and two 2200 b.hp. units, making a total of 9400 b.hp.

For continuous operation the Diesel engine is the equal of any prime mover. A number of records from the above plant show continuous twenty-four hour service over periods of several months.

One of the most interesting records received was that of a 1250-b.hp. unit carrying practically full load, operating for a non-stop period of 78 days. The remarkable feature of this run was that the engine was located at an elevation of 4000 ft. and carried practically a full load at sea level rating. I found records of 30 days continuous operation common, the units then being shut down for only a few minutes for inspection and minor adjustments.

In marine service, where the strains and stresses are severe, records show that the Diesel engines stand up even better than modern steam-driven units. They are proving their ability to this type of service every day and there is little doubt that in time they will supplant steam.

For such work, direct connection to generators which in turn furnish power to the motor-driven propellers makes an ideal arrangement. The ease with which the vessel may be handled and the absence of standby losses while docked are factors which cannot be overlooked.

Again, the space usually taken by the boiler and coal bunkers can be used for cargo, thus greatly increasing the carrying capacity of the vessel. In most cases the fuel oil is carried in the false bottom or bilge tanks and is easily pumped from one compartment into another, thereby keeping the vessel "trimmed" at all times.

Two large vessels recently built for operation on the Great Lakes are equipped in this manner, electricity being used also for heating, lighting and cooking. Controls are lcoated in the pilot house and on the bridge where the movement of the boat is at all times under the direct control of the captain or officer on watch. The illustration shows a 350 b.hp. Diesel-electric plant in a modern ocean-going tug boat.

Several railroads have adopted this system and in many sections switching is done by "oil-electric" locomotives. Although the first cost of this type of locomotive

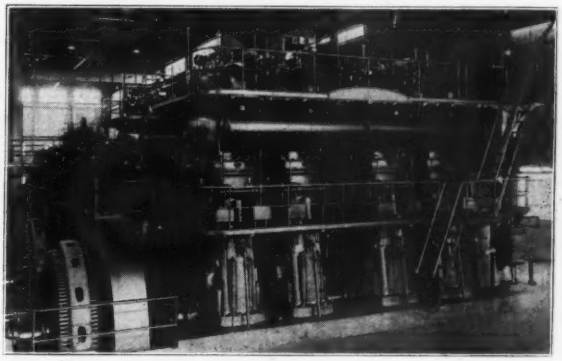


FIG. 1. DIESEL PLANT OF MOCTEZUMA COPPER CO. CONTAINS TWO 2200-HP. AND FOUR 1250-HP. ENGINES

is somewhat greater than the usual steam driven unit, the cost of operation is so low that it is bound to receive consideration.

In locomotives of this type a Diesel engine drives a generator, which in turn furnishes power for motors geared to the driving axles. Such a combination not only achieves the advantage in economy of the oil engine but also offers the opportunity of realizing the favorable operating advantages of the electric locomotive in locations where complete electrification is not advisable due to infrequency of traffic or other causes.

Some of the unique features of a locomotive of this type are worthy of mention. The engine uses what is known as the solid fuel injection. That is, fuel is injected by a plunger pump, thus doing away with high-pressure air compression, air piping and pumps. The generator is specially designed with a separate shunt field and differential series field. The result is that the generator output automatically adjusts itself to the load demand.

On such a locomotive, the control consists of a throttle lever and a master controller which gives reverse and series-parallel connections for the motors. The speed of the engine and therefore its power is controlled by a throttle lever similar to that of a steam engine. Opening the throttle increases the speed from idling to full speed within the limits of the overspeed governor. The compounding of the generator field adjusts the output of the generator to correspond to the power set by the throttle opening. The speed of the locomotive can be varied by changing the throttle and during acceleration the locomotive automatically accelerates at a rate determined by this throttle opening and without any loss of power in the rheostat. These locomotives are operating on a guaranteed fuel consumption of not more than 0.45 lb. of oil per b.hp. hr.

Before a meeting of the A. I. E. E., A. H. Babcock, electrical engineer for the Southern Pacific R. R. showed that by extremely careful tests the best fuel efficiency obtainable with an oil-fired locomotive is 6.4 per cent, the average seldom exceeding 5.5 per cent. This means that of the heat units obtainable from a pound of fuel, only about 6 per cent are delivered to the locomotive driving wheels. This being the case, a locomotive of the oil-engine electric type could run four times as far with the same amount of fuel as a steam oil-fired locomotive. The cost of an oil-electric locomotive is approximately twice that of a steam locomotive, but the former can operate nearly 24 hr. per day. The possibility of doing twice the work of a steam locomotive means that the distance between division points may be increased to 500 mi., greatly decreasing the number of roundhouses. Since little water is used, water stations may also be decreased, which means considerable in many sections of the country.

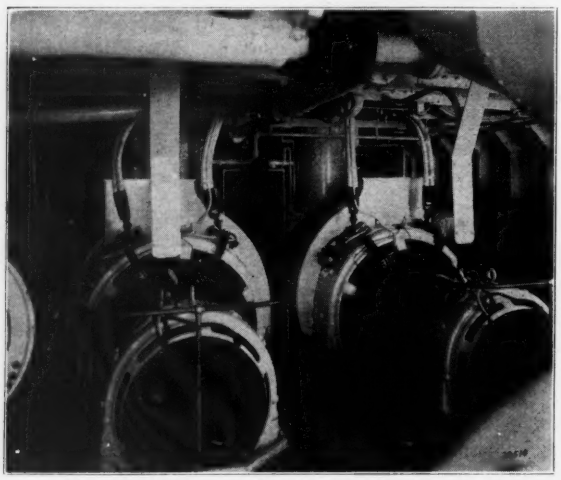


FIG. 2. DIESEL ENGINES ON SEA-GOING TUG DRIVE GENERATORS

When these facts are considered, they afford ample reason why our supply of fuel oil should be used in internal combustion engines instead of under steam boilers as is done on the Western coast. The era of cheap fuel is past and no longer should we waste, as we do, our most valuable remaining resource. The writer believes the time is not far distant when coal distillation plants will be erected in the coal regions and will lead to a more economical solution of the fuel problems of our country.

Two large, low-temperature coal distillation plants are now being erected and, from tests, the normal figures are approximately as follows: One ton of ordinary coal containing about 35 per cent volatile matter will produce the following:

Gas	5000 cu. ft.
Heavy fuel oil	18 gal.
Gasoline or motor spirit	3 gal.
Sulphate of ammonia	15 lb.
Tar	62 lb.

If one ton of coal were burned under a boiler it should produce 200 b.hp. hr. (figuring 10 lb. of coal per b.hp. hr). This same amount of coal distilled, the by-products being used under boilers and in Diesel engines, would develop practically 800 b.hp. hr. or four times the power we could obtain if the fuel was burned in the usual manner.

In addition to the products enumerated, used in the production of power, we should have remaining (approximately) 41 lb. of lubricating oil, 2 lb. of wax, 15 lb. of ammonia sulphate and a residue which is used in the manufacture of artificial leather.

When the above process becomes universal, regarding which there is little doubt, coal distillation will furnish fuel for Diesel type engines, cheap gas for heating, cheap electric power, ammonia for fertilizer and smokeless cities will be the result.

Power Co. Effects Savings by Two Innovations

By C. W. Geiger

WHEN PULLING power and electric light cables through conduits under the streets of San Francisco, the Great Western Power Co. now uses a low body trailer on the bed of which special heavy supports have been built, enabling as many as three reels of cables to be carried.

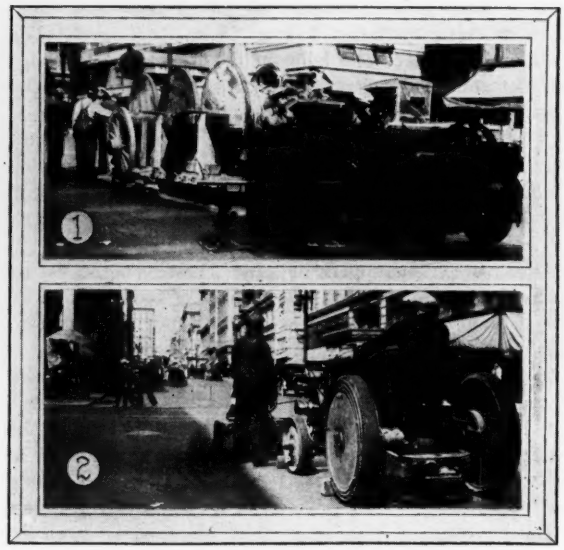
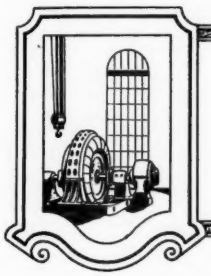


FIG. 1. PULLING THREE CABLES SIMULTANEOUSLY BY MEANS OF LOW BODY TRAILER
FIG. 2. THIS TRACTOR IS USED TO PULL CABLES THROUGH CONDUIT

Any one or all three of the reels can be unwound simultaneously without removing them from the special supports, thereby saving considerable time and cost, as previously it was necessary to roll the reel onto the ground near the manhole and then jack the reel up on two jacks so that the cable could be unwound. The old method of jacking the reel up is shown in the foreground in Fig. 1.

Another labor saving device is the use of the tractor (used for towing the low bed trailer) in pulling cables through conduits under the street. Previously, this was done by means of a hand operated winch which required the services of four men and required from a half an hour to nearly an hour to pull a cable a distance of one block. Through the use of the drum on the tractor as shown in Fig. 2, one or several cables can be pulled a distance of a block in about 10 min., eliminating the four men previously required for operating the hand winch.



ELECTRICAL MACHINERY FOR LIGHT AND POWER

Operation of Converting Apparatus---VI

TRANSFORMERS, THEIR THEORY, OPERATION AND METHODS OF CONNECTING. BY V. E. JOHNSON

PERHAPS THE ONLY reason that alternating current has become so widely used is because of its easy transformation from one voltage to another. Since this transformation is effected through the use of the alternating-current transformer, it is apparent that this piece of equipment is one of the most important of all electrical devices. It is also one of the most efficient of all machines since it transforms energy without the employment of any moving parts.

Figure 1A shows schematically the elementary portions of a transformer. It consists essentially of three elements; the primary winding which receives electrical energy at primary voltage; a secondary winding, and a magnetic circuit. The secondary has no electrical connection with the primary. The magnetic consists of an iron circuit which serves as the connecting link between the two windings. These three parts correspond roughly to the driving pulley, belt and driven pulley in Fig. 1B.

In an ideal transformer, the ratio of transformation is equal to the ratio of the number of primary turns to the secondary turns.

Thus if in Fig. 1A the primary coils have 1000 turns, and the secondary 100, the ratio would be 10 to 1. If 1000 v. were applied to the primary terminals, a potential of 100 v. would be set up across the secondary. In an actual transformer this ratio would vary to some extent when load is applied, but at no load it is very nearly correct. Applying this to Fig. 1A the analogy can be carried further. The relative speeds of the driver and driven is inversely proportional to their diameters. For a given "driver" r.p.m. the driven pulley can rotate at any desired speed by merely making it of the proper dimensions. This ratio is quite accurate at no load but when loads are applied, the speed of the driven pulley drops due to slippage, just as the secondary voltage drops under load due to resistance and reactance drop.

TRANSFORMER AT NO LOAD

When voltage is applied to the primary terminals of a transformer a current will flow which will set up a magnetic field in the iron circuit. The amount of current which will pass is determined by the following factors:

a. The resistance of the coil.

b. The back voltage set up in the coil by the action of the magnetic flux. This is usually spoken of as counter electromotive force, c.e.m.f. Without getting deeply involved in the mathematics of the relations involved it can be seen that with a given resistance, sufficient current will flow to raise the flux value to a point where c.e.m.f. plus resistance drop equals Impressed Voltage.

With an iron circuit arranged as in Fig. 1A the opposition to the flow of magnetic flux is exceedingly slight,

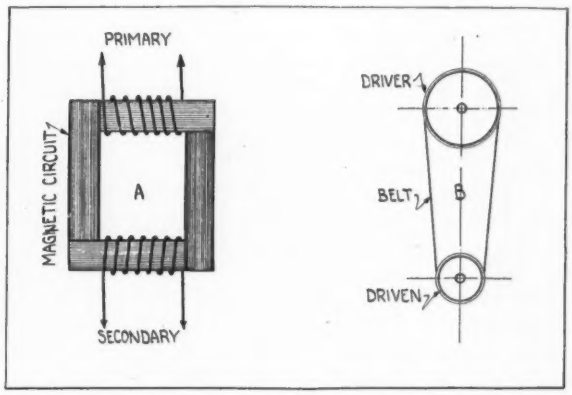


FIG. 1. DIAGRAM OF SIMPLE TRANSFORMER TOGETHER WITH BELT AND PULLEY ANALOGY

due to the circuit being complete, so that the amount of current required to reach the point of equilibrium is extremely slight; or using the ordinary terms "the magnetizing current is small." This small current would of course produce only a small resistance drop, so that practically the c.e.m.f. would equal the impressed voltage.

The formula for determining the relation between the factors above is as follows:

$$\text{c.e.m.f.} = \frac{4.44\ N\ T\ f}{10^8}$$

where N equals the number of magnetic lines of force
T equals the number of turns in the coils
f equals the frequency in cycles per sec.

With a negligible resistance drop the above formula also covers the impressed voltage. In other words—other conditions being unchanged—the flux flowing in the magnetic circuit is proportional to the number of primary turns.

By using the same formula it can be seen that the secondary voltage is proportional to the number of turns and the flux. Eliminating a number of variables, we may say:

$$\frac{\text{Primary voltage}}{\text{Secondary voltage}} = \frac{\text{Primary turns}}{\text{Secondary turns}}$$

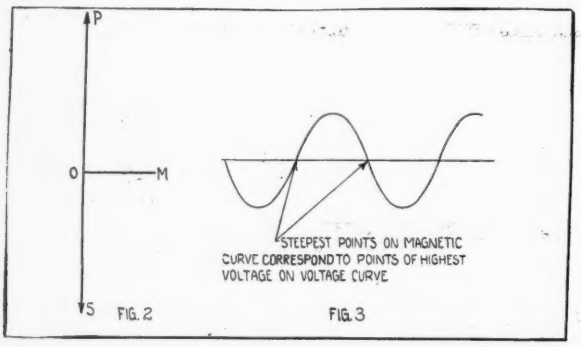


FIG. 2. SIMPLE VECTOR SHOWING PHASE RELATIONS OF CURRENTS IN A TRANSFORMER

FIG. 3. REPRESENTATION OF THE MAGNETIZING CURRENT OF A TRANSFORMER

We have already applied this formula to a practical example.

Phase Relations at No Load

In the preceding articles on alternating current theory it was shown that if a voltage be applied to a magnetic coil having no losses, the current flow would be 90 deg. behind the pressure. Vectorially this would be represented by Fig. 2, where Om is the magnetizing current and OP the impressed voltage. If this magnetic coil be the primary of a transformer, having a secondary coil a voltage would be set up in the latter at an angular position OS in Fig. 2. The truth of this can be seen from Fig. 3. The sine wave in the latter represents the magnetizing current Om and the magnetic flux which is in phase with it. The voltage set up in the secondary varies with the rate of change of flux, and as this is maximum at the zero point of the curve as indicated it follows that the peak of the secondary voltage wave is 90 deg. from the peak of the current wave, and in direct opposition to the primary. The angle POm can be 90 deg. only on the assumption that there are no losses of any kind, so that the energy input is zero.

Figure 2 also shows graphically the rule which states in effect that the flux set up in a circuit by an alternating current always flows in such a direction that the induced voltage opposes the impressed voltage. This was found to be true for the c.e.m.f. and is equally true for the secondary voltage, which after all is set up by the same flux.

Transformer Under Load—(Non-Inductive)

In an ideal transformer having no losses the input and output would be equal, or for a non-inductive load

$$Ip \times Ep = Is \times Es$$

where I and E are the currents and voltages and the letter p or s indicate primary and secondary respectively. From this we find that

$$\frac{Ip}{Is} = \frac{Es}{Ep} = \frac{\text{Turns s}}{\text{Turns p}} \qquad (1)$$

The flux set up by the current taken from the secondary opposes the flux generated by the load current in the primary, and just cancels it, so that in an ideal transformer the net quantity of magnetization remaining is that due to the original primary magnetizing current m. This is deduced mathematically as follows:

Magnetic effect of secondary current equals

$$Is \times Ts$$

where Ts indicates the number of secondary turns.

The magnetic effect of the primary current equals

$$Ip \times Tp$$

where Tp indicates the primary turns.

The total current in the primary is Ip — M.

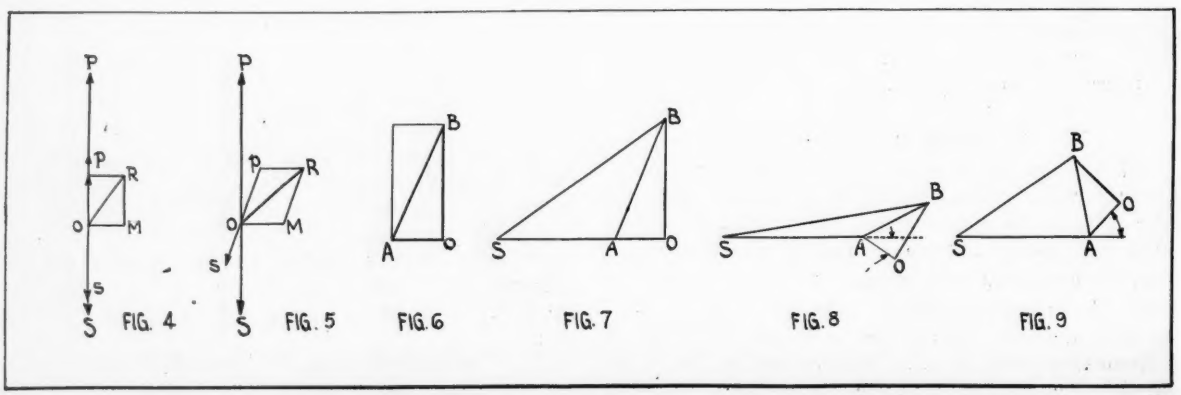
From (1) above IsTs = IpTp. Hence the magnetic flux at all loads is proportional to m.

The vector relations in an ideal transformer under non-inductive load conditions is shown in Fig. 4. Here OP is the primary voltage, Om the magnetizing current and OS the secondary voltage. If the voltage OS be applied to a non-inductive load, a current will flow, and this will be in phase with the secondary voltage. This current is represented by the current vector Os. The current Op will flow in the primary, which will be in exact opposition to Os and of a value $\frac{Ts}{Tp}$. This will be the load component, and combined with the magnetizing current Om will form a resultant OR which constitutes the actual current flowing in the primary coils.

If the load increases, the angle POR will decrease, and the power factor will become better. This accounts for the common statement that "lightly-loaded transformers make the power factor low."

Transformer Under Load (Inductive)

If the load on the transformer consists of motors or other apparatus in which the current taken from the line lags behind the voltage, the conditions resulting would be



FIGS. 4-9. VECTORS AND IMPEDANCE DIAGRAMS SHOWING ACTION OF THE TRANSFORMER

as shown in the vector diagram in Fig. 5. Here OP is the primary voltage, OS the secondary voltage, and Om the magnetizing current. The secondary load current lags behind the voltage an angular distance sOS, and is represented by the vector Os. The primary current which flows as a result of this is Op. This combines with the magnetizing current m to form the resultant actual primary current OR—which is the value indicated on an ammeter connected in the primary leads. It can be seen that OR lags behind OP a greater amount in Fig. 5 than it does in Fig. 4.

Transformer Under Actual Conditions

In the previous paragraphs the transformer has been considered as being ideal, and as having no internal drop under load conditions. In actual practice, however, there are several factors which materially affect its performance. A transformer, being a reactive circuit, can have its internal voltages represented by a vector diagram as Fig. 6. When current is taken from its windings there is an ohmic drop AO, in phase with the current, and a reactive drop OB, at right angles to it. The resultant of these two is the vector AB which represents the impedance drop, and is analogous to the relations previously shown to exist in a generator. The triangle AOB is called an "impedance triangle" and by using the units of length to represent percentages of normal voltage, can be made to include the drops of both the primary and secondary windings. That is to say, if there is a 2-per cent primary drop, and a 3-per cent secondary the total vector would be drawn to represent 5 per cent.

In order to apply this impedance triangle to the finding of the regulation of a transformer under varying load and power factor conditions, a diagram as in Fig. 7 is used. Here SA is the full load secondary voltage, drawn to represent 100 per cent, while AOB is the impedance triangle of Fig. 6. SB then represents the secondary voltage which must be generated. If the power factor be other than unity, the line SA will not coincide with AO but will be at an angle as in Fig. 8 which shows a lagging current, or as in Fig. 9 which is for a leading current. In the latter two figures xAO represents the angular displacement between current and voltage.

The manufacturers supply data on the reactance and impedance of transformers, and with this information it is easy to construct an impedance diagram such as that in the previous four figures. It is possible to determine from this just what the terminal conditions will be under any load.

Development of Electrical Equipment in 1924*

Progress in Design and Construction of Electrical Apparatus Shown in All Fields. New 110,000-v. Underground Cable a Notable Achievement

ALTHOUGH THE total volume of sales for the electrical industry during the year was somewhat below that of 1923, for certain important classes of apparatus, including central station equipment, automatic stations, traffic signalling, street lighting and industrial heating, it exceeded all previous records.

In addition to the production of steam turbine generators of large unit capacity there has been an increase in the use of higher steam temperatures and pressures. The installation of the 65,000-kv.a. waterwheel generators at Niagara was completed and they still represent the maximum in generator unit capacity and physical dimensions.

Both the unit capacity and the number of installations of automatic station equipment for railroad, hydroelectric, central station and industrial service were increased as compared with previous years and there were numerous additions to the line of relays which rendered possible this great economic advance in operating practice.

The growing necessity for high tension underground conductors was responsible for intensive developmental work not only on conductor cables and their insulation but also on the terminals, junction boxes, etc. necessary for a complete dependable underground transmission system.

Research work with the lighting generator was continued and valuable data was secured in regard to the protection of transmission lines. A new portable X-ray outfit was produced and the manufacture of clear fused quartz on a commercial basis was achieved.

While the electrical apparatus referred to in this article are all products of the General Electric Co., the reference to their development will serve as an indication of the tendencies in design and construction as well as the general trend of progress in the electrical manufacturing industry as a whole.

Cable Developments

The continual increase in the volume of electric power which must be transmitted in congested districts has served to emphasize the importance of development work on paper insulated, oil filled underground cables for higher tensions.

An extensive program of development was actively pursued in the laboratories and factory. In order to supplement this work by subjecting the cable to the voltage fluctuations which occur in actual operation, a field test was conducted with the cooperation of the Adirondack Power

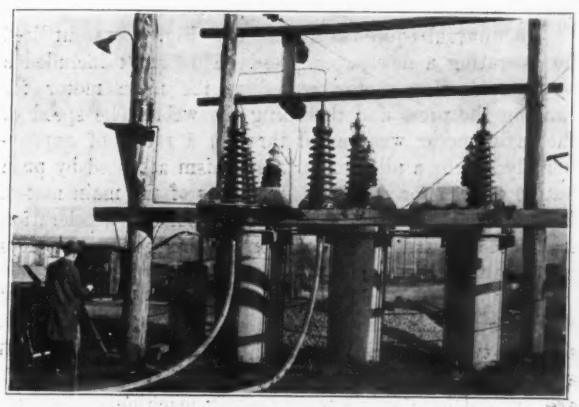


FIG. 1. FIELD TEST OF 110,000-V. SINGLE CONDUCTOR PAPER INSULATED CABLE

*Abstract of article, "Some Developments in the Electrical Industry During 1924," by John Liston in the General Electric Review for January, 1925.

& Light Corp. on its 110,000-v. circuit at North Albany, N. Y. A commercial length of experimental single conductor cable was put under continuous test on this line early in October. The development of this cable has been paralleled by that of suitable cable joints and terminals for voltages up to 132,000.

Automatic Stations

There were no radical changes in the design of automatic switching equipments. The tendency during 1924 was to produce unit equipments which are easily handled and installed, and to standardize as much as possible. Automatically controlled hydroelectric generating equipments have made it possible to develop economically the small isolated water power site as they have eliminated the necessity for attendants, thereby decreasing the operating cost. Many companies are taking advantage of this fact and are installing small automatics to operate in conjunction with their large manually operated stations, while others are developing large power sites and installing large capacity multiple unit automatic stations. A

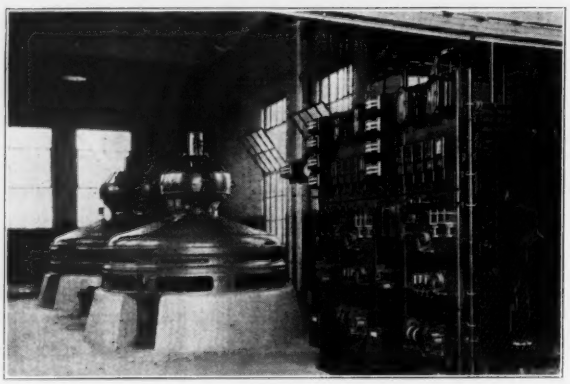


FIG. 2. TYPICAL SMALL (492 KV.A.) HYDROELECTRIC STATION WITH AUTOMATIC CONTROL

typical small hydroelectric station with automatic control is shown in Fig. 2.

Motor Applications

In the field of motors, there were many applications of recently developed types to new fields. The alternating current brush-shifting adjustable speed motor was utilized for the first time for the operation of stoker drives in boiler plants.

An unusual equipment, shown in Fig. 3, was utilized for operating a newspaper press. This outfit included a separate starting motor geared to the main motor for starting the press and threading the web. The speed of the main motor was varied through a range of approximately 3:1 by a pilot motor mechanism actuated by push-button control for shifting the brushes of the main motor. In addition to securing remarkably smooth acceleration and speed control, the operation of this equipment does not involve rheostatic losses.

Small drawn shell type motors of exceptional speed were provided for the operation of woodworking machinery. A typical application of this character consists of 5 small motors with operating speeds of 25,200 r.p.m. which are utilized by a single woodworking machine.

Among the special types of motors produced, the motor shown in Fig. 4, together with its control equipment, is of interest. These motors were developed for operation in the presence of vapors from gasoline, ether, etc. They were only constructed in ½-hp. and 5-hp. sizes, but the exhaustive tests which they passed with entire success indicate that the construction of industrial type motors can be modified so as to render them suitable for efficient operation under gaseous atmospheric conditions.

The line of single-phase repulsion induction motors utilizing a squirrel cage rotor construction and eliminating centrifugal switch, was extended to include the odd frequency motors and the reversing type of motor. The reversing type is capable of reversal from full speed in one direction to full speed in the opposite direction, such as for hoist and crane service, but requires special control in the case of elevators.

Single-phase motor development also included a line of varying speed repulsion motors designed primarily for direct connection to slow speed exhaust fans. These motors are totally enclosed and have the series characteristics desirable for fan load.

A new line of synchronous motors for driving air and ammonia compressors employ a double bar squirrel cage winding which permits the designing engineers to have greater control of the current and torque during the starting period of the motor, and in addition permits the motor to be thrown on at full voltage with the assurance of minimum starting current and proper torques.

Industrial Motor Control

Work along the lines of completely enclosing all types of motor starters and speed controllers was practically completed.

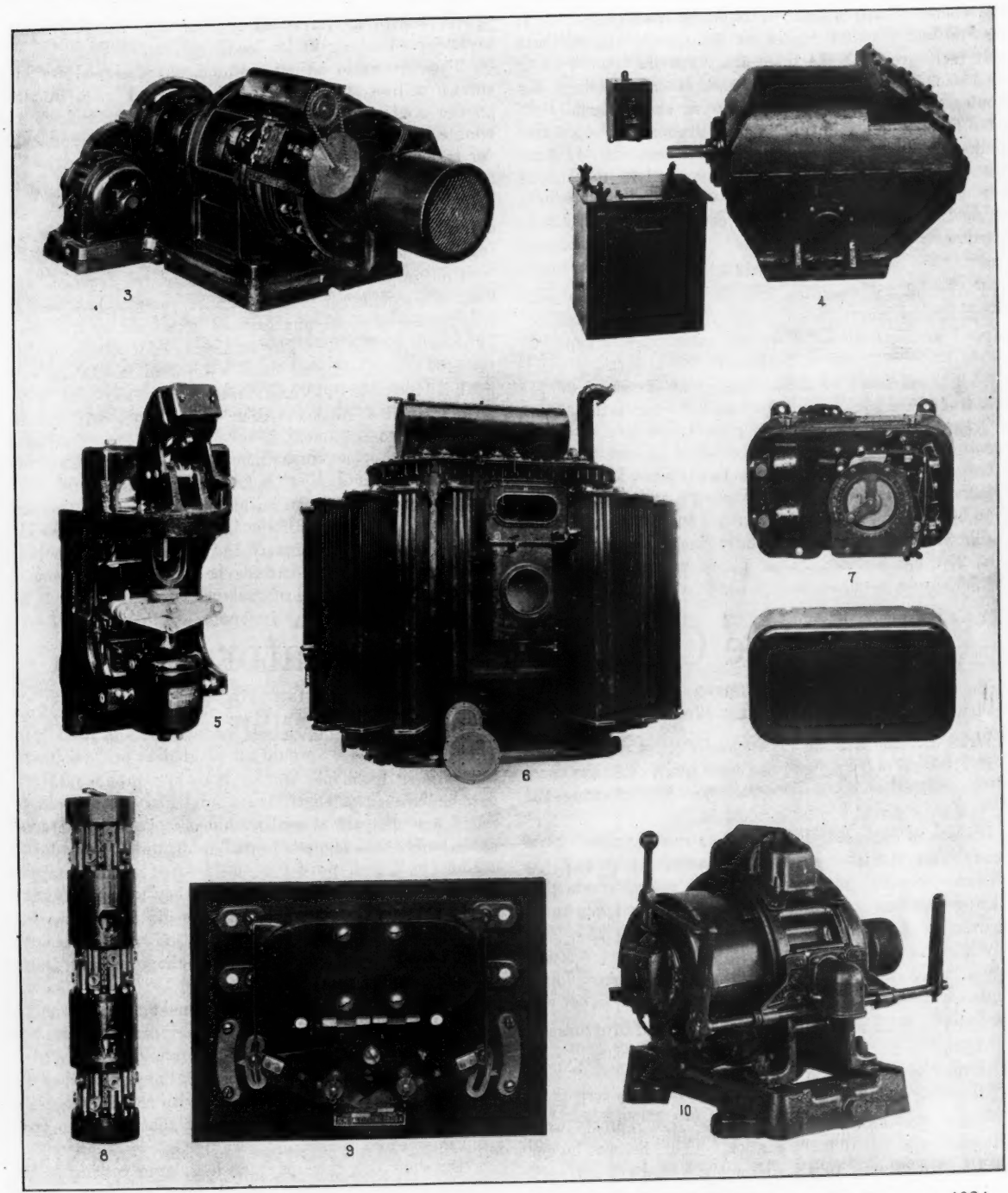
Heretofore, fractional horsepower motors have, to a large extent, utilized standard wiring device switches, which were primarily designed for lighting circuits. A new design of single-pole and double-pole enclosed tumbler switchers, assembled in boxes with conduit wiring, for throwing small motors on the line, now gives these motors a class of control equipment comparable with that provided for large motors. These new switches are given a definite horsepower rating.

In Fig. 7 is shown a definite time relay which was developed to provide the accelerating period. It consists essentially of a motor-driven train of gears and magnetic clutch, and switching mechanism to provide for either opening or closing the contacts at the end of the time for which the device is set. It is adjustable from a few seconds up to several minutes, and is adaptable to many applications where a definite time adjustment is needed.

A new thermal overload relay, Fig. 9, was designed to follow the heating and cooling curve of the average induction motor, and is particularly adapted to service where it is important that the motor be permitted to carry heavy short time overloads intermittently without being tripped out by the overload device. It will permit the motor to do any kind of work below a safe-operating temperature.

Switching Apparatus

Radical improvements were made in the design of motor-operated mechanisms for oil circuit breakers in which a small vertical type motor operates over wide variations of voltage both on alternating-current and direct-current circuits. It has a series characteristic which enables the mechanism to operate speedily and with great force at the beginning of the closing stroke. The



FIGS. 3—10. TYPICAL APPARATUS WHICH MARKS THE PROGRESS OF THE ELECTRICAL INDUSTRY DURING 1924

Fig. 3. Double Motor Printing Press Equipment Which Gives Remarkably Smooth Acceleration. Fig. 4. For Use in Atmospheres Containing Explosive Vapor, This Gasoline Fume Proof Motor and Starting Equipment Was Developed. Fig. 5. Centrifugal Motor Mechanism for Oil Switch Operation. Fig. 6. Transformer with Compartments for Bringing Cables Into Unit Without Exposing Terminals. Fig. 7. Definite Time Relay for Providing Accelerating Period. Fig. 8. Ten Point Crab Type Ratio Adjuster. Fig. 9. Temperature Ratio. Fig. 10. Brush Shifting Motor with Base, Pulley, Bracket, Rocker Shaft, Levers and Handle.

fly ball linkage, driven by this motor is so constructed as to give a cushioning effect at the end of the closing cycle and the tripping device is designed to trip free in case the breaker is closed on a short circuit.

The entire mechanism is small, Fig. 5, simply constructed but very positive in its operation and it is capable of operating any of the panel mounted oil circuit breakers. A similar but larger mechanism was designed for breakers up to and including the 154,000-v. size.

A triple-pole combined disconnecting switch and fuse was produced for 15,000, 25,000 and 37,000-v. circuits. It is designed so that the blowing of any one of the three

fuse elements will actuate a tripping mechanism which opens all three switch blades simultaneously and prevents their reclosure until the fuses are renewed.

A complete new line of connectors and fittings for tubular conductors simplified the work and materially reduces the cost of installing outdoor station bus structures as all soldering and sweating joints have been entirely eliminated. They will carry the same current as the tubing to which they are connected, and will withstand any mechanical stresses to which they will ordinarily be subjected.

Transformers

Early in the year the two largest three-phase self-cooled transformers ever constructed were completed, shipped and installed. They are rated 15,000 kv.a., 60 cycles, 72,000-132,000 v. with a 12,470-v. tertiary winding. The efficiency of these units is in excess of 99 per cent from half load to full load and the regulation is 1.1 per cent. Other 15,000 kv.a. units are now under construction. These units are indicative of the rapid growth in capacity which has characterized self-cooled transformer constructions since 1920 when a maximum rating of 10,000 kv.a. was first attained.

The larger water-cooled units included two 20,000 kv.a., 25-cycle, 3-phase, 22,250-11,500-v. and four 20,000-kv.a., 60-cycle, 3-phase, 69,000-13,800-v. transformers, all provided with an improved type of tap changing equipment for switching under load.

The new ratio adjuster which was designed for this service utilizes standard contact fingers, Fig. 8, in place of the contact rollers previously used, but retains the simple and compact drum construction which is applicable for transformers of any type, size or voltage.

In air blast transformers, the maximum in capacity was increased by the construction of an 18,500-kv.a., 25-cycle, 3-phase, 11,800-2950-v. unit arranged for switching under load by means of reactors and switches.

One improvement in transformer construction consists of a scheme for bringing underground cables into the transformer without any exposed terminals or live parts, as shown in Fig. 6. The underground cable is brought up into a compartment filled with fluid insulating compound. Suitable links and connectors are provided in this compartment so that the cable may be properly "phased in." This compartment is connected by means of bushings, to an upper compartment which is kept filled with oil by means of a pipe leading to the main tank. The transformer high voltage bushings pass into this compartment through the side of the tank and links are provided so that the transformer may be disconnected to facilitate testing of the cable. Access to both compartments is attained by the removal of gasketed covers.

The Champion Prevaricator

A Visitor on the Job Tells of the Big Engine He Erected When He Was on the Road. By George H. Wallace

I WAS erecting a little high speed Corliss engine down in Missouri one time and one noon hour, after the usual mid day exercise, we were sitting around "chewing the rag," when a casual visitor strolled in.

He was of the usual type that has run "ingines" 7 yr. in this place, 19 in another, 14 somewhere else, and has also "steamboated" a dozen years or so more, in addition to being foreman in a stone quarry, while he looks to be not over 35 at the most.

Without ceremony he strolled over to where I was sitting on the back cylinder head placed conveniently on a couple of blocks, and remarked:

"I've run the biggest ingine in the State of Muzzurah!"

"Yeah?"

"I sure hev."

"How big was it?"

"Two—hundred—and—fifty—hoss" he replied, with particular emphasis to make it stick. "Yep, hit war bigger than that little toy o' yourn. How big is that ere ingine?"

"Oh—about 400 hp. or so—if it gets right down to business."

"Four hundred hoss! You cain't tell me thet."

"Well, there's the name plate on the generator which says '250 kw.' and you can figure it out for yourself."

"Thet stuff don't mean nothin' t' me! Thanks," he replied when I tendered a pretty fair cigar, "don't care if I do."

"Yeh, I uster be a 'rectin' ingineer' myself; put in 10 yr. at it. 'D'ja ever hear of th' big ingine thet I 'rected over in the steel districk?"

"Can't say that I have. Got a little time before the whistle blows and we would all be glad to hear of it. Go ahead."

"Well suh," he began as he scratched a match on the leg of a seedy pair of well wrinkled trousers, "hit were," (puff puff), "th' biggest," (puff, puff, puff), "gol darned ingine (puff, puff p-u-f-f p—u—f—f—f) "that you ever see, fer a fact. Hit war so big that they wasn't a foundry thet could cast the frames and if they did cast them, there was no way to get 'em down to the job without draggin' them on the ground. So they cast them ere big frames right on the job.

"They got the pattern makers to make the patterns and then cut them up into pieces and shipped them in nine box cars down to the job, where 50 moulders shoveled in sand with a steam shovel for a month. Then they got inside and chopped the pattern out with axes. One of the laborers saved enough lumber out of that wreck to build a house of seven rooms and a garage.

"But how did they pour such a large casting to say nothing of two of them?" asked John Nein, one of the engineers.

"I knew you would ask thet question. First we thought of puttin' up a furnace right on the job, but it would take such a big one that we could not spare the time. So we did the next best thing; we melted the iron over in Skelpville an ran a special train of 25 cars loaded with melted iron. When we came through Cobbleville, the heat of the melted iron set fire to the depot and dozen freight cars, and by the time the sheriff found out about it, we were 50 mi. down the track going 60 mi. an hr.

"We got the melted iron there all right with no other damage than to burn up two wooden bridges, but they held

till the train got over. We ran that ere train right into the power house—in one side and out the other—and as we passed the moulds, we dumped the melted iron in on the fly.

"By gum, we made a good guess for we didn't have 50 lb. left over when we got done. Close shave. Hit took them ere castin's a month to cool off. Then we had a job borin' them out and facin' up th' bearings, but I rigged up a borin' bar made out of an old 14-in. gun that I borrowed from the navy yard and bored them ere bearin's out as slick as a whistle.

"Fer facin' off th' back ends of th' frames I borrowed a big planer that was used durin' th' war to plane the decks of battleships, set it up on edge and fastened the tools to the bed of th' planer. Worked all right too.

"We were 7 days melting the babbit fer th' bearin's. Had a battery of 13 hog kettles set up out in the yard, and carried them in with a steam crane and dumped them into the bearin's. We did the best we could with th' babbitin' and then put a few men in with adzes to chip off a few high spots. Did a good job too—old ship carpenters.

"Th' rest of th' stuff came from the factory. The wheel was a big problem but I surmounted it in fine style. They made the wheel and shaft all right, but after they had it made, they had no way to ship it, it was so gol durned big. Th' sup'intendent called me up by phone and asked my advice.

" 'Thet's easy, sez I. Thet's *dead* easy. 'Well, if you kin do it, come on up here on th' first train and show us; I'm frum Muzzurah.' So I went up. They had th' wheel on th' shaft and standin' up against th' side of a cliff back of th' shop.

"I got a couple of flats that th' Gov'ment uses to haul torpedo boats inter drydock with, built a bearin' on each car and rolled th' wheel in between the two cars, then jacked th' track up till th' wheel was jist about runnin' on th' ground.

"Then I hired th' whole danged railroad fer a couple of days, with double tracks, and with four of these ere,—wot you callem—mallet locomotives to pull each car and started out. But first one ingine was ahead and then tother and we tore up a sight o' track till I hit on the scheme o' hitchin' all th' throttles together with a bar and let one ingineer handle the whole eight ingines.

"We got on bully. Covered some 200 mi. th' first day. Sort of laid out the mail trains some, but,—we was a-payin' for hit."

"But how did you get around the bridges?" asked Martin O'Conner. "A wheel that size would not go through the bridges and the tunnels. How did you work it?"

"Jes' as you said, we went around 'em. Well, we got that ere wheel down on the job all right only at one place where the telegraft poles was a leetle too close to th' track, the cranks knocked over about a couple o' miles o' them and tangled th' telegrams all up.

"I built a track,—a double one,—right in between them ere bearin's and ran that ere train in till th' shaft war right over th' spot. Then we washed th' dirt from in under th' cars with a fire hose and th' cars settled down nicely till th' shaft war in place.

"Th' car with each of th' cross heads on we ran right up to th' end of th' frames, and pulled 'em in with six span o' mules."

"How did you get the mules out?" asked Fred Sanders.

"How did we get 'em out? W'y we unhitched 'em and drove em out of an oil hole.

"T' keep th' dirt out o' the crankpins, we slipped one of th' hog kettles we melted th' babbit in, over each one and they were a good fit too. T' git th' c'nectin' rods on we filled th' frames with a steam shovel and then rolled the rods in on rollers, like movin' a house, and then dug th' dirt out again.

"Th' c'linders we could not load on cars, so we put wheels under them and brung 'em down that way till we come to a tunnel that war too small to let 'em through. I war about stumped fer awhile till I happened to see a steamer with a tow on the river nearby and I rented th' whole danged outfit.

"Took up th' track and relaid it down to th' river and ran th' c'linders onter th' two barges and we were all fixed, only that we had to go down th' Wabash river, to th' Ohio, and up that to th' Swampskahatchet Run, and up that to within half a mile of th' plant.

"There I had to dredge a canal up to the plant, and in a week we had them ere c'linders bolted to th' frames."

"How many bolts did you use, and how big were they?" asked Earl Young. "Must have been big bolts."

"Yeh, they war. They war 29 o' them fer each front head and they war 8 in. in diameter and 17 ft. long."

"But why use 29,—an odd number?" piped in Doyle, the Master Mechanic.

"W-e-l-l, 28 *might* have done, but—I like to be on th' safe side. I mout add that we ran out o' material on th' front heads, and when we come to put th' back heads in, we welded 'em in.

"But, as I was a-sayin' when we come to line her up, I used a clothesline, and a 10-ft. pole with spikes in th' ends fer trams. She come pritty good too, only out about half an inch, and that don't matter with a big ingine.

"Each o' th' valves war a leetle over 3 ft. through and they come one on a car. Th' dashpots were made out o' old chubular b'ilers cut in two, and babbited and th' valve gear rods war made out o' old well drills.

"Th' eccentrics war 12 ft. in diameter and 2 ft. wide, and fer crab steels we cut up a few old anvils. Worked fine too.

"One day when we had bin runnin' a week or so, I sent four men inter th' c'linder to tighten up th' follower bolt nuts with a span o' mules, and some smart Aleck opened th' bypass and started up, and do you know, them ere men and mules come out o' hit in pretty good shape 'xceptin' thet one o' th' mules got t' kickin' and cast a shoe.

"But what became of the men when the piston came back against them," asked Bert Ellis.

"Oh, they hid in th' clearance. One o' th' whippletrees got jammed inter th' indicator pipin' and had ter be bored out with a auger.

"But, you say, this smart Alec opened the bypass; how big was th' bypass?" asked Jack Romack.

"Th' bypass war 12 in. and th' main throttle war 4 ft. But as I was a-sayin', we war a little excited about th' condenser and especially when we were pullin' up 48 in. of waccum. With a big ingine you kin git a powerful lot of waccum.

"This ere canal that we floated th' c'linders in on was turned inter a supply and we pumped th 'whole danged

Swampskahatchet River as dry as a Kansas barn raisin'. Got a dredge and deepened it till th' water of th' Ohio flowed up hill enough t' git us that ere 48 in. o' waccum.

"How many boilers did it take to generate steam for that engine?" asked Clarence Chamberlain.

"Well suh, I don't remember *jist* how many, but th' war a couple o' three acres o' them at least. Anyway when th' air pump started to git in its work, the discharge flowed down th' main street o' th' town and washed out two blocks o' pavin' brick, a movie theatre, three stores, a livery stable and a Greek restaurant; it did, fer a fact.

"But how did you oil an engine the size of that?" asked Andy Kramer. "Must have taken a lot of oil."

"Yep, it did, a right smart o' hit. Mostly we used coal scuttles and had tanks o' ile all around th' ingine like water barruls in a lumber yard. Some o' th' ingineers got a new wrinkle and used waterin' pots with sprinkler drilled out a leetle bigger. Say, you may think that ere ingine war big, but yuh kain't begin ter realize it till I tell yuh that th' governor war so allfired high up, that we had ter use a 30-ft. 'xtenshun ladder ter get to hit. After a week o' clombin' th' ladder and dodgin' governor balls th' size o' a pork barrul, the boys sort o' got tired tryin' t' hit th' ile holes with a dipper every time th' danged thing cum past 'em, and they rigged up a garden hose, and took turns a-holdin' it.

"How many engineers did it take to run that engine," asked Bob Turner. "Must have taken a lot of them."

"Yep, hit sure did, but I don't remember just how many. Y' see, I never saw all of 'em at once for while they war about a dozen men on a shift, there war only a few o' them in sight. They all had roller skates and it took an hour t' skate around th' ingine and I never had time t' count 'em all.

"Ever have any accidents with that engine?" asked Jack Potts.

"No. None t' speak of. O—h, she lost her waccum one day and when she barked out doors, th' ixhaust knocked a corner off th' office and pushed th' warehouse across th' yard 17 ft.

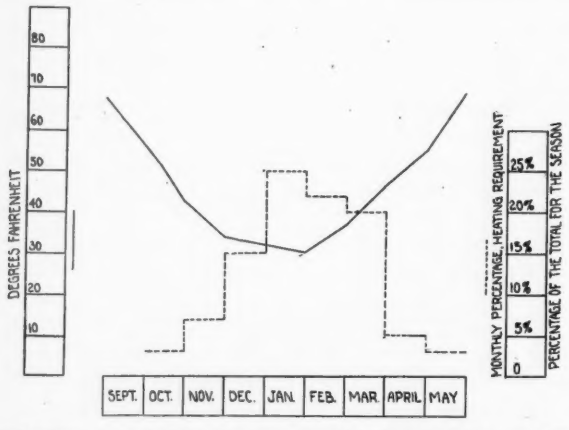
"How big was that engine? Ever have an indicator on it?" asked Tom Lawrie.

"Well suh, I kain't tell yuh. I don't know; nobody knows. We never could get a indicator big enough ter tell much about hit. I finally made a indicator with a drum made outen a piece o' 18-in. pipe, but th' danged rope hitched to th' crosshead kep a breakin' an' they wa'n't enough smokestack cable left over t' reach, so I give it up as a bad job. But boys, she sure war a hummer," he concluded as the one-o'clock whistle blew, and with a bland "thanks" for another cigar, he started off up the track for the next town.

Economic Consideration of Power Service

TANGIBLE AND INTANGIBLE COSTS DETERMINE CHOICE BETWEEN CENTRAL STATION AND ISOLATED PLANT SERVICE. BY JOHN D. MORGAN

IN READING an article some time ago on scientific management, I noted one paragraph somewhat as follows: "It may be possible, after a careful examination and investigation, to effect considerable economy by purchasing current for the power needs of the plant and scrapping the engine-room equipment." With this idea in view, let us give a careful study to the practicability and possibility of central station current replacing the isolated plant.



HEATING REQUIREMENTS FOR SMALL INDUSTRIAL PLANT. FULL LINE, TEMPERATURES; BROKEN LINE, HEATING LOAD

In order to bring out the main points in the discussion, let us take a specific example of a foundry and machine shop, located in New York State, doing an annual business of about $1,000,000 and employing about 400 men, with an invested capital of $1,200,000. Such a plant would have a useful horsepower consumption of 550 with an average load factor of about 25 per cent.

It is obvious, by referring to the chart, that it will be necessary to provide heating during a considerable portion of the year. A plant of the size under discussion would have about 20,000 sq. ft. of radiation, which would mean that the season's steam supply would be from 8,000,000 to 10,000,000 lb. of steam, and would require a 100-hp. boiler using about 400 T. of coal for the heating season.

In taking up the power plant for this size plant, there are five practical installations for the industrial plant. These five are as follows:

1st. Steam engine, line shaft drive.

2nd. Steam engine direct connected to electric generator, individual motor-driven equipment.

3rd. Steam turbine, direct connected to electric generator, individual motor-driven equipment.

4th. Oil engine, line shaft drive.

5th. Oil engine direct connected to electric generator, individual motor-driven equipment.

The first and fourth installation systems are obsolete and it is on these power installation systems that central station current is by far the most economical, due to the fact that no man will install electric service and still continue to use line shafting and belting, because of the power loss in line shafting and belts, which often reach a total of 60 per cent of the total power developed, and will average well over 30 per cent. With the bulk of evidence in favor of motor-driven equipment, the only fair comparison between central station energy and industrial plant energy

TABLE I. APPROXIMATE COST SUMMARY OF ENGINE PLANT, 600 KW., 882 B.HP.

	Total Cost	Unit Cost per Kw.	Unit Cost per B.H.P.
3-294-hp.water tube boilers,set in place,hand fired	$11966.40	$ 19.95	$12.28
3-200-kw.engine generator units complete with engines, d.c.generators, jet condensers & equipment	29691.16	49.48	33.65
3-steam separators	297.00	0.495	0.337
1-closed feed water heater	684.00	1.14	0.775
2-feed water pumps and equipment	581.50	0.969	0.66
2-steam traps	83.10	0.138	0.094
2-feed water injectors	173.40	0.289	0.1965
Station gages	62.55	0.104	0.071
Damper regulator	121.94	0.202	0.138
Breeching	1235.40	2.06	1.40
2-3-in. water meters	170.00	0.283	0.192
Steel stack	2669.00	4.45	3.02
Coal handling trestle	663.18	1.11	0.75
1-10-ton coal scale	230.00	0.384	0.26
Engine oiling system	154.08	0.257	0.175
Piping,valves,hangers,pipe covering	6773.41	11.29	7.68
7-panel switchboard	1246.81	2.07	1.42
2-2-ton hoists	100.00	0.17	0.11
Conduit ducts,insulators,cable and busses	408.00	0.675	0.46
Tools	141.00	0.23	0.16
Total equipment	57448.93	95.74	65.10
Total building	15660.00	26.10	17.78
Total building and equipment	73108.93	121.84	82.88

TABLE II. APPROXIMATE COST SUMMARY OF TURBINE PLANT, 600 KW., 1000 B.HP.

	Total Cost	Unit Cost per Kw.	Unit Cost per B.H.P.
2-500-Hp. stoker fired boiler and superheater	$26500.00	$ 44.16	$26.50
1-600-kw. turbo generator with jet cond. and pumps and misc. equipment	13066.92	21.77	13.07
1-25-kw. turbo generator set	1509.07	2.51	1.51
2-feed water pumps	840.48	1.40	0.84
2-heating service pumps	330.77	0.55	0.33
1-1000-hp. closed feed water heater	904.00	1.51	0.90
2-steam traps	88.00	0.16	0.09
Station gages	288.65	0.48	0.29
Damper regulator	150.00	0.25	0.15
Steam flow meter	247.80	0.41	0.25
Oil filter	68.50	0.11	0.07
2-3-in. water meters	170.00	0.26	0.17
Breeching	1215.40	2.02	1.21
Coal trestle and concrete coal bunker	2469.20	4.11	2.47
1-crane	808.00	1.34	0.81
Piping	3015.18	5.02	3.01
Stack	2669.00	4.45	2.67
1-10-ton coal scale	230.00	0.38	0.23
Generator air duct	112.50	0.18	0.11
5-panel switchboard, electrical control conduit, ducts,insulators,cable and busses	1538.15	2.56	1.54
Tools	275.00	0.45	0.28
Total equipment	56496.88	94.16	56.50
Total building	14500.00	24.16	14.50
Total building and equipment	70996.88	118.32	70.99

is made when both plants supply current for motor-driven equipment.

Taking up installation plan No. 2, that of a direct-connected steam engine installation, this plan no doubt covers the majority of old small industrial plants. A list of the equipment and building, as well as the approximate present day installation costs, is given in Table I. This table is made up for a three-unit station and allows ample reserve capacity.

The more modern of the small industrial plants is covered by plan No. 3, that of a direct-connected turbine generating plant. Table II is given to show the list of equipment for this type of isolated plant as well as the cost distribution.

The cost of a plant on the plan of No. 5 is variable; however, it should be possible to put in a complete 600-kw. Diesel engine set with electrical and general station equipment for $58,500 with a building cost of $11,150.

In order to compare these three plans, it is necessary to refer to the fact that in plan No. 2 and plan No. 3, that ample boiler horsepower has been figured in to take care of the demand for steam for heating, while in plan No. 5, no such provision has been made.

In considering the cost of power from any plant, it is necessary to take into account the tangible and intangible costs. The tangible costs are:

1st. Operating costs:

Total operating and maintenance labor.

Total operating supplies, coal, water, lubricants.

Total maintenance material.

2nd. Supervision from salaried employees proportioned to charges based on capital invested in former plant.

3rd. Fixed charges:

Interest on capital invested.....6 per cent

Insurance .....1 per cent

Depreciation building, 2 per cent;

equipment .....5 per cent

Taxes .....1 per cent

With these data we are now in a position to consider the costs, assuming that, when operating ten hours per day for 300 days per year, the total output for the year would be about 1,110,000 kw.-hr. With this assumption, Table III shows the normal costs of current for the various plans mentioned.

Reference to Table III will show that when the heating load is taken into consideration, the isolated plant of

TABLE III. COMPARATIVE GENERATING COSTS

Item	Engine Driven Plant Hand Fired Boilers	Cost per Kw.-H. Cents	Turbine Driven Plant Stoker Fired Boilers	Cost per Kw.H.	Diesel Engine Driven Plant	Cost per Kw.H.	100 H.P. Steam Heating Plant	Central Station Current
Operating and main. labor	$ 5800.00	.52	$ 4350.00	.39	$ 3300.00	.29	$2000.00	
Operating supplies	11622.00	1.04	9960.00	.90	9906.00	.90	2200.00	
Maintenance materials	759.00	.07	634.00	.06	521.00	.047	150.00	
Total operation	18181.00	1.635	14944.00	1.34	13727.00	1.24	4350.00	
Fixed charges	8719.00	.78	8503.00	.77	8491.00	.77	775.00	
Charge for factory supervision	500.00	.045	500.00	.045	500.00	.045	250.00	
Total cost	27400.00	2.46	23947.00	2.155	22718.00	2.055	5375.00	$24541.00
Cost per kw.-hr.,power and heat	2.46¢		2.155 ¢					
Cost per kw.-hr., power alone					2.055 ¢			2.21 ¢
Cost per ton of coal	5.00		$ 5.00		6¢ per gal.		5.00	
Water rate	31.2 lb.		27.2 lb.					
Coal rate	4.04lb.		3.50 lb.		1.1# of oil			
Kw.-hr. generated	1,110,000		1,110,000		1,110,000			1,110,000
Credit for heat furnished	$ 4300.00		$ 4300.00		$ 5375.00			$ 5375.00
Power cost less heat credit	23100.00		19649.00					
Total power cost plus heat credit					28093.00			29916.00
Actual cost per kw.-hr., power alone	2.08 ¢		1.78 ¢		2.055 ¢			2.21 ¢
Actual cost per kw.-hr. for power and heat	2.46 ¢		2.155 ¢		2.62 ¢			2.70 ¢

In the credit for heat furnished, a credit of only 80 per cent is given to the engine and turbine plant, due to the small amount of coal used to heat buildings at night.

The central station rate is based on the following data:

Hp. demand charge of $16.20 per year.

3c for first 200,000 kw-hr. 2c for next 200,000 kw-hr.

2½c for next 200,000 kw-hr. 1¾c for next 400,000 kw-hr.

1½c for all amounts over 1,000,000 kw-hr.

the size under discussion can generate current at a more reasonable cost than the central station can furnish.

The costs given in Table III are those of a tangible form; however, the intangible costs should be taken into account which cannot be figured in dollars and cents. The main items in this class are as follows:

1st. The uncertainty of power costs of the isolated plant versus the contract price of central station current.

2nd. The charges resulting from power interruption.

3rd. Charges for engineering advice for the maintaining of high efficiency in the isolated plant.

4th. Charges resulting from the difference in the amount of money earned by the capital stock tied up in the power plant.

5th. Worry of plant officials.

Let us consider the fact of the intangible costs; the uncertainty of power costs of the isolated plant is no greater than central station service for, once an industry does away with its isolated power plant, it is at the mercy of the central station and the costs of the latter vary as those of the small plant do.

Charges resulting from power interruption are not likely to be greater in the isolated plant than with central station service in one year. One central station had six station failures, 29 sub-station failures, and 113 line and cable failures, a total of 148 failures in one year. The small plant should not have any more than this in view of the fact that it usually has at least 12 hr. per day to fix up and at least one full day a week of shut down to make repairs. The power plant can earn at least 10 per cent on the money invested over and above the 6 per cent interest charge on the investment. The worry of the plant officials is a matter that is hard to discuss; if the officials favor either the plant or the central station, the actual worry can be greatly magnified in either direction.

Many people will say that a plant smaller than that discussed will show up in favor of central station current. In this regard, it is my personal opinion that if the connected load is less than 200 kw., the central station rate will be lower. In the matter of coal economy, however, the best of the small isolated plants can hardly show a yearly average coal rate of better than 3.50 lb. per kw.-hr. when the coal used for banking is taken into account. It is now quite a common matter for a well operated central station to show an average yearly coal rate of 1.45 or an actual saving of 2.05 lb. of coal for each kw. generated.

Rankine Cycle Applied to Modern Practice

EXTRACTION OF STEAM FOR FEED WATER HEATING AND REHEATING STEAM COMPLICATE THE DEFINITION OF A STANDARD OF COMPARISON BETWEEN UNITS

SINCE THE TIME that the Rankine cycle was first established as a standard for the comparison of steam engine efficiencies, there has been developed the practice of heating the feed water by live steam abstracted from the engine. This practice has resulted in an increase in economy but when it is used the Rankine cycle as formerly defined, does not apply. The practice has also been adopted of late years of reheating the steam after a partial expansion, a process which can be repeated one or more times after which the steam is expanded to the back pressure; further, these feed-heating and reheating cycles can be combined, and the question arises, can a standard of comparison be defined that will meet all cases? This question has been analyzed by H. Riall Sankey and his conclusions appeared in a recent issue of Engineering, London.

It has been suggested that if, in the original definition, the sentence: "Beyond the point expansion takes place adiabatically, doing work until the pressure in the cylinder is equal to the back pressure against which the engine is working," be extended, the definition would apply to all cases. The point referred to is the point of cutoff at which the supply of heat from the boiler ceases.

In order to make the sentence just quoted meet the new conditions imposed it is proposed that it be changed to read as follows: "Beyond that point expansion takes place adiabatically, doing work until any desired pressure is reached, when the steam is reheated at constant pressure to any desired temperature whence adiabatic expansion takes place; such reheating process can be repeated as often as desired. Finally, the steam is expanded adiabatically until the temperature of saturation corresponding to the boiler pressure is met. From this point either adiabatic expansion continues doing work until the pressure in the cylinder is equal to the back pressure against which the engine is working or the steam is expanded so that a portion of the heat is transferred to the feedwater in such a manner that at every temperature the amount of heat so transferred is equal to the increase in water heat at that temperature until the pressure in the cylinder is equal to the back pressure against which the engine is working."

In practice, the following main cases occur resulting from the omission of one or more of the factors of this comprehensive definition:

(1) Saturated steam with simple adiabatic expansion to the back pressure

(2) Superheated steam with simple adiabatic expansion to the back pressure

(3) Superheated steam with one or more reheats with final adiabatic expansion to the back pressure

(4) Saturated or superheated steam with feed-heating expansion to the back pressure

(5) Superheated steam with one or more reheats and final feed-heating expansion to the back pressure.

DEFINITION OF THERMAL EFFICIENCY

Thermal efficiency of any heat engine is defined as the ratio of the heat utilized as work to the heat supplied and in the case of the standard of comparison just defined these two heat quantities can be determined by formulas but even in the simplest cases these formulas are cumbersome and the calculations are lengthy.

The definition of the Rankine cycle as adopted by a previous committee but amended to accord with the modern developments referred to in this paper, is given in full as follows: It is assumed that all the component parts of the steam plant are perfect and that there are no losses due to initial condensation, leakage, radiation, or conduction and that there is no clearance in the cylinder. The feed water required is taken into the boiler at the exhaust temperature and its temperature is gradually

raised until that corresponding to saturated steam is reached. Steam is then formed at constant pressure until dry saturated steam is produced after which, if the steam is to be superheated, heat is added at constant pressure and at increasing temperature, until the required temperature of superheat is reached. The steam is introduced into the cylinder at constant pressure, displacing the piston and performing external work equal to the absolute pressure multiplied by the volume swept through by the piston up to the point of cutoff.

Beyond that point expansion takes place adiabatically, doing work until any desired pressure is reached, when the steam is reheated at constant pressure to any desired temperature whence adiabatic expansion takes place; such reheating process can be repeated as often as desired. Finally, the steam is expanded adiabatically until the temperature of saturation corresponding to the boiler pressure is met. From this point either adiabatic expansion continues doing work until the pressure in the cylinder is equal to the back pressure against which the engine is working, or the steam is expanded adiabatically until the temperature of saturation corresponding to the boiler pressure is met. From this point either adiabatic expansion continues doing work until the pressure in the cylinder is equal to the back pressure against which the engine is working, or the steam is expanded so that a portion of the heat is transferred to the feed water in such a manner that at every temperature the amount of heat so transferred is equal to the increase of heat in the water at that temperature until the pressure in the cylinder is equal to the back pressure against which the engine is working. The steam is then completely exhausted from the cylinder at constant pressure corresponding with the lower limit of temperature, work being done on the steam by the engine during exhaust, equal to the absolute back pressure multiplied by the total volume swept through by the piston. The steam is thus removed from the cylinder and the cycle is complete.

Foreign Developments in the Engineering World

New Type of Water Meter, Sensitive Hydraulic Valve and New Horsepower Formula Apply Familiar Principles in New Way. By J. H. Blakey

IN THE water meter of the "Imperial" type manufactured by the firm of Beck & Co. of London, England, an attachment has been provided for the distribution of a determined and regulable quantity of water or other liquid. This appliance thus becomes a distributor-meter which can be used for the mixture of fluids in any required proportions, or for the discharge at short periods of a given amount of liquid. The meter proper is composed of three groups of cylinders, each containing a piston, arranged symmetrically around the vertical central shaft of the bell cover, of which only one is shown in the accompanying cut. The piston rods *t* are connected to an arm which is a prolongation of a bronze cap forming a valve moving on a spherical seat. This seat, which is of vulcanite in a water meter, has three ports communicating with the spaces below the pistons; a fourth port *l* in the center of the seat leads to the discharge. The water from the feed pipe, which is not shown in the figure, passes into the upper part of the cover and exerts an equal pressure on all the pistons; the space under one of these pistons will be connected with the discharge by means of one of the ports of the valve, as shown, and this piston will be depressed by the entering fluid. At the same time the other pistons are raised in sequence and the water flows into the spaces below them in consequence of the arrangement of the ports in the cap and in the seat. These pistons are then successively lowered and drive the water below them out through the discharge. The circular motion thus given to the cap is transmitted by a shaft to the registering mechanism above.

To this registering mechanism is attached a distributor, consisting of a horizontal shaft *a* carried by a bracket and connecting by gearing. A graduated dial *c* with a movable stop *r* is attached to the bracket. Mounted on the shaft in front of the dial is a friction clutch *k*, of which the male part carries a radial index *i*, so arranged that the rotation of the shaft brings it in contact with a vertical lever V which is fixed to the stem of the valve *s* which operates the discharge. This valve being closed, the movable stop on the dial is clamped at a point corresponding to the discharge desired, and by pushing the male part of the clutch, the index is turned clock-wise to bring it into contact with the lever. The discharge valve is then

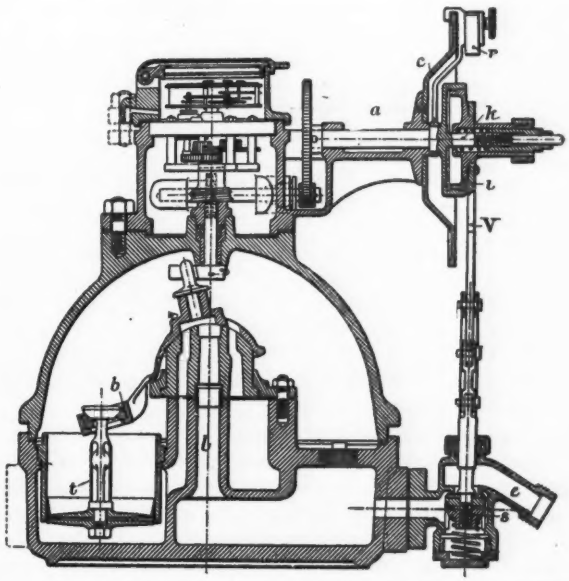


FIG. 1. REGISTERING WATER METER DISCHARGES FIXED AMOUNT AT REGULAR INTERVALS

opened by means of a hand lever; the flow of water turns the index in the other direction until the stop comes again in contact with the valve lever, and the discharge is thus automatically closed. It will be seen that to change the period of discharge it will be necessary to change the gearing connecting the registering mechanism with the horizontal shaft. The manufacturers claim that the distribution of a liquid is by this means determined with an error not greater than one per cent.

M. Christian Leu has made a number of experiments in the mixing of mineral oils, which he describes in a

recent number of the Moniteur du Petrole Roumain. The experiments were made for the purpose of finding a formula which will give the viscosity resulting from a mixture of two or several oils of known viscosity, or which will determine the volume of two or several oils to be used in making a mixture having a required viscosity. The formula is

$$[P(V-V^1)\ (V+v)]\div[2p(V^1-v)\ (V-v)] = 2.718{,}28$$

in which V is the Engler viscosity of the heavy oil and v that of the light oil; V^1 is the resulting viscosity of the mixture, P the percentage of heavy oil and p that of the light oil. The Engler coefficient of viscosity is the time of outflow through a given orifice of 200 c.c. of the oil under investigation, at the test temperature, divided by the time of outflow, through the same orifice, of 200 c.c. of water at a temperature of 20 deg. C. (68 deg. F.). This latter quantity is called the water constant.

The Homeyard High Pressure Hydraulic Valve

At the recent exhibition at Wembley, England, there was shown a new type of hydraulic valve, made by Glen-

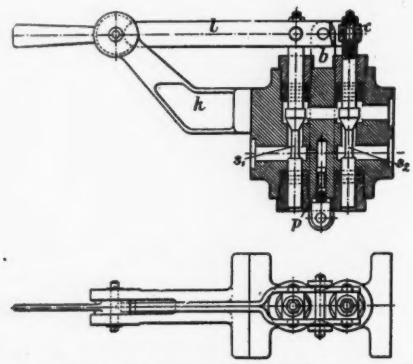


FIG. 2. BALANCING OF PISTON PRESSURES RENDERS VALVE UNUSUALLY SENSITIVE

field & Kennedy of Kilmarnock, Scotland, which can be used for hydraulic pressures up to 230 atmospheres. This valve is so sensitive that the smaller models, used with pressures up to 60 atmospheres, can be operated by the touch of a finger. This smaller type is composed of two valves equilibrated side by side; one for admission s_1 and the other for escape s_2, each terminated by a cross c pivoted upon the fork of a lever l. Between the two crosses is arranged a transverse axis to which are suspended two plates b passing on each side of the casting and reunited below on the head of a small piston-plunger p. The pressure of admission acts always on the upper face of this plunger and so keeps both valves closed. At the extremity of a bracket k is a handle with a groove, into which fits the rounded end of the lever. Thus the raising of the handle opens the discharge and its lowering opens the admission valve and in the mean position the two valves are closed. This model is to be used only with single acting cylinders; for double-acting cylinders there is another model in which each valve has two ports.

Industrial Uses of Liquid Combustibles

In the May 3 issue of the Zeitschrift des Vereins Deutscher Ingenieure is devoted entirely to articles on combustible liquids. Here are found communications made by various engineers to the Hamburg section of the Union of German Engineers. The communication of M. Schlawe, the director-general of the petroliferous enterprises of the Disconto Gessellschaft en Roumanie, is the most important, and it is continued in the May 10 issue. This article is largely a history of the development of the petroleum industry in the old and the new worlds and is filled with figures and statistical tables. In conclusion, the author examines the possibility of replacing petroleum by other oils, such as schist and paraffin oils, oils from the distillation of lignite, etc. He insists especially on the value of these latter, which are now much used in Germany and in which he sees for his country the means of replacing, at least in large part, the importations of petroleum and its by-products.

New Formula for Calculating Horsepower of an Engine

At the meeting of the French Academy of Sciences on July 16, a note from M. M. E. Tournier was read, giving a formula for condensing engines which is somewhat different from those in use in this country. The author claims that the error will not exceed one per cent. The formula is as follows:

$$\text{Actual horsepower} = [13.6M(W \times F)]\div[(1+L) \times (2.3+p)]$$

in which M is the maximum efficiency of the engine, which is a constant; W is the weight of dry steam discharged by the engine, in unit time, in kilograms; F is the adiabatic fall of heat of the steam from the absolute boiler pressure to that of the condenser, in calories; L is the percentage of steam which does no actual work, and p is the condenser pressure in kilograms per square centimeter. (A French calorie is equal to 3.968 B.t.u. and a kilogram is 2.2 lb.) In the complete note, which is to be found in the Comptes Rendus of the Academy, the author shows the methods he has used to determine the different elements of the formula.

A Poor Water Power Development

By G. H. Kimball

THERE IS MUCH talk in various parts of the country about developing every available bit of water power. In some instances the power obtained will amount to less than 500 hp. at times of medium high water. In this connection it is pertinent to say to all who contemplate any small development, that they see that all rights pertaining both to the waterpower site and any necessary right of way for the transmission lines.

In the accompanying sketch is shown what was done by one company in building a power station and preparing the necessary dams to use the water. The course of the river was as shown, leaving the power station out of it. The dam for No. 1 mill made a head of about 25 ft. available and since this mill was for sale at the time, it would have been an excellent place for a station. Its steam plant had two good brick chimneys so that with the addition of new water wheels, engines and boilers, together with the necessary generators, no buildings would have been needed and there would have been considerable space for other purposes. This whole property could have been purchased for $30,000.

Mills 2 and 3 had about 15 ft. head on their water wheels and could have been purchased for about $50,000.

Above these mills the banks were high and largely of rock formation. If these two mills has been demolished and a dam built high enough to keep the pond at the height of the tail water from No. 1 mill, a head of 33 ft. would have been possible and a power house could have been constructed in level ground below the dam with the water wheels on the same level as the generators, making direct connection possible. The storage would have been sufficient to have carried the entire load at that time for at least eight hours.

The above propositions were perfectly feasible and required no unusual engineering features. Now let us see what was finally done. The sketch is quite plain and it will be seen that the water was induced to run to one side of the river by a low wall that was laid across the river on the rocky bed. At the start, the head gate house was located wrong since, in times of high water, all of the water that was used in the plant and that which was wasted

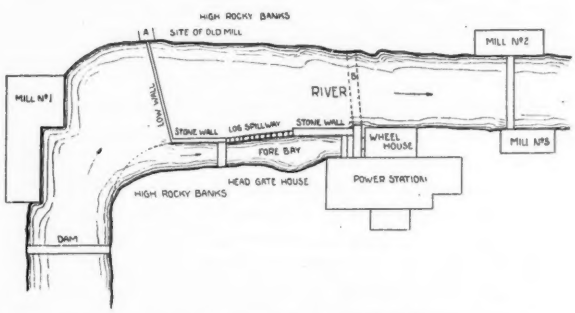


DIAGRAM SHOWING DETAILS OF SMALL WATER POWER DEVELOPMENT

over the spillway was obliged to run under the gates. This often contained parts of trees, or any other rubbish that the high waters picked up. It was quite usual for these gates to get partially obstructed with these obstacles. In view of this, the head gates should have been placed closer to the rack and inside of the spillway.

After building this plant for developing power a greater mistake was made. To gain a footing on the opposite river bank, about 8 ft. was purchased from the owners of No. 1 mill. Then it was found that there had been a dam across the river at this point to supply power on the opposite bank. Having once been used for power this made the site a water privilege and the owner thereof felt that he should be paid damages for the water being turned to the other side of the river away from his site. All of the land from the low wall to a point opposite the new power station wheelhouse could have been purchased for $2000 and the old water power rights would have been included. The constructing company, who also acquired all property necessary, acted upon poor legal advice and ignored the claims of the owner of the opposite water site and labored under the impression that their low cross wall was all that was necessary. It was later brought out that water cannot be turned from its natural course and that by so doing they had destroyed the value of the old water privilege. After the power company had taken over the completed property, the owner brought suit for recovery of damages. If the case was ever settled, it is probable that he received considerably more than would have purchased the site in the first place but it required more than that for the power company to defend the case in addition to the amount of trouble that was caused.

It will be seen that the arrangement of the canal and forebay allowed no storage and it has been proven many times that after the mills shut off their wheels so that no water ran down the river for a longer or shorter time, that one of the 400 hp. wheels in the power station would empty the canal in five minutes. How much better it would have been to have bought the land opposite and continued the stone wall as shown by the dotted line B and had a storage that would have been capable of giving 400 hp. for at least an hour. If this had been done, much coal would have been saved in running steam engines for short periods ranging from 15 min. to one and one-half hours.

Rate of Boiler Corrosion Determined by Simple Tests

CORROSION often occurs in boilers fed with distilled water, which does not form scale but does contain dissolved oxygen. Such corrosion occurs at points in the boiler where there are eddies or dead ends and while it is almost impossible to predetermine the places where the worst of it will occur, it is sometimes possible to get some idea, from certain symptoms, as to what actually takes place in the boiler, says T. R. McDermet in Powerfax.

Red water from gage cocks or drum is one of the most serious symptoms, indicating, in some cases, sufficient corrosion to attack all the wetted surfaces. On the contrary, however, it may indicate feed pipe corrosion, which may occur without corrosion in the boilers.

Internal inspection is the best guide in the case of drum type boilers. A small amount of corrosion may show in the drum, but excessive corrosion will show by the pitting of tube sheets, rivet heads and the tube beads projecting into the drum after rolling. Free use of a geologist's hammer is recommended, to clean the surfaces so that they can be inspected. If it is desired to prevent the corrosion from spreading, small areas of the drum can be cleaned with grinding wheels and revolving wire brushes and brought to a high polish. These areas can be marked off with a chisel so that they can be identified when the boiler is next opened.

Without excessive work, it is almost impossible to arrive at a knowledge of the condition of boiler surfaces that are covered with a mixture of iron rust and scale-forming materials. If the boiler has been operated in such a condition for a long period, the use of deaerating equipment will often cause the boiler surfaces to free themselves from this deposit. This is because the solution of iron in deaerated water takes place in the ferrous form, while aerated water dissolves the iron in ferrous from but immediately oxidizes it to ferric. The rust found deposited on boiler surfaces is usually in the ferric form. Deaerated water holds in solution the products of corrosion, whereas aerated water precipitates them as a protective coating. Air-free water, over a period of time, will eventually dissolve and loosen these iron oxides, although complete solution is impossible.

What happens is that after deaeration has been used for about 9 mo. the pitted places begin to be covered with a black powdered oxide which can be removed by brushing. At this point the operator usually becomes alarmed that

the boilers are damaged, but as a matter of fact, they have just reached the point where inspection is possible. Now is the time to polish a portion of the sheet, and observe its action during the next run. The small amount of oxide that will form—just enough on the polished surface so it can be wiped off with a rag—will probably reassure him. Even this oxidation is probably caused by admission of aerated water when filling the boiler.

In header types of boilers, it is obviously impossible to make a careful inspection of inside of header surfaces. An indication of the amount of corrosion can be obtained by polishing the header caps opposite three or four badly corroded tube ends. After polishing, these header caps will not corrode during the next run at the same rate as the tubes in which corrosion is already started, but an approximation of the rate of future corrosion can be made.

Polished plates in the drum also will give an indication of the rate of boiler corrosion. Such plates may be about 2 in. by 6 in. of ¼ in. boiler steel of about the same composition as the sheet itself. Three pairs of plates, at equally spaced intervals in the drum, will usually suffice. One plate of each pair should be suspended by a wire and the other should be in metallic contact with the sheet. The reason for this is that the suspended plate generally corrodes less than the boiler surfaces and the fixed plate usually corrodes at a more rapid rate, so that from inspection of the two plates, it is possible to get a fair idea of the rate of corrosion. It is to be expected that the polished plates will show a slight black deposit, after a run, but it is desirable that they be free from pit holes.

Air Compressor Cylinder Lubrication

IN THE OPERATION of air compressors, most of the troubles experienced can be traced to the use of a poor grade of oil in the air cylinders. The importance of good air cylinder oil is now recognized by all the leading manufacturers of lubricants and all of the reputable oil companies today put out an oil which they recommend for air cylinder service, therefore the recommendation of the compressed air society is to use only the best grade of air cylinder oil as sold by any of the reputable oil companies.

Use only enough oil to furnish lubrication, that is, only enough to keep the cylinder walls and valves just greasy, neither dry nor wet. A small quantity of the correct oil will maintain the necessary oil film whose prime function is to assure a piston seal. With the correct high grade oil in use the presence of any deposits on the valves means only one of two things, either too much oil or the intake air is dirty.

As a guide to show approximately the amount of air cylinder oil required for good lubrication, the accompanying table shows the drops per minute for different sized air cylinders and also the maximum quantity in pints per 10 hr. that should be used. These figures are for one cylinder only. Cylinders having Corliss valves should use 50 per cent more oil.

If the oil you are using in your air cylinder does not give proper lubrication when used in the above amounts, consult the oil company from whom you are buying your oil. They have experts who follow up lubrication troubles and will see that you get proper oil. If lubrication troubles exist, make sure you also examine the air intake to your compressor which can be done by getting into the air intake and seeing if dirt adheres to the sides and pockets, for if your intake air is dirty this would help to account for your trouble and a proper air cleaner should at once be installed.

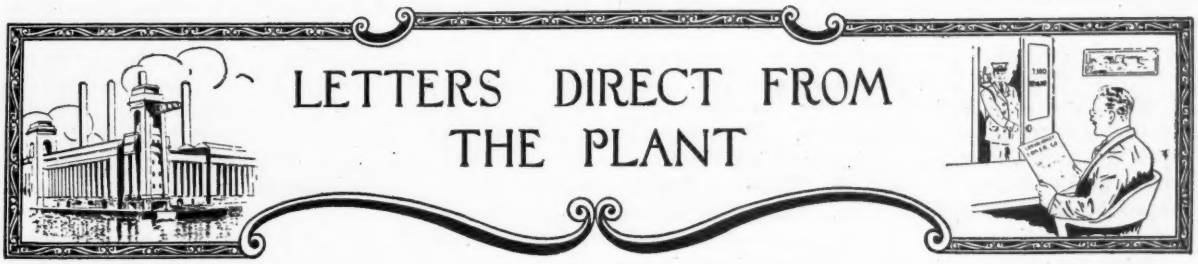
OIL REQUIRED FOR AIR COMPRESSOR CYLINDERS

Cylinder Diameter	Piston Displacement	Rubbing Surface Square Feet	Oil Feed—Drops per Minute	Oil Feed—Pints per 10 Hours
Up to 6″	Up to 65	Up to 500	1	.05
6″ to 8″	65-125	500-750	1	.08
8″ to 10″	125-225	750-1100	1	.11
10″ to 12″	225-350	1100-1500	1-2	.14
12″ to 15″	350-600	1500-2000	2-3	.20
15″ to 18″	600-1000	2000-2600	3-4	.27
18″ to 24″	1000-1800	2600-3600	4-5	.36
24″ to 30″	1800-3000	3600-4800	5-6	.48
30″ to 36″	3000-4500	4800-6000	6-8	.60
36″ to 42″	4500-6500	6000-7500	8-10	.74
42″ to 48″	6500-9000	7500-9000	10-12	.90

Never clean cylinders with kerosene, benzine or gasoline. This is dangerous and should be prohibited. If there is any tendency for carbon to form in a machine, the use of soap suds is recommended. A good cleaning solution is made of one part of soft soap to fifteen parts water. These suds should take the place of oil for a few hours and be fed into the air cylinders about once a week either by means of a hand-pump or through the regular lubricator at a rate about 10 times as rapid as that of the oil. The cleanliness of the air valves when inspected, as they should be periodically, will indicate whether greater or lesser application of the soap suds should be made. After using soap suds, open the drain cock of the air receiver and of the inter-cooler in the case of compound machines, to draw off any accumulated liquid. Oil should be used again for one-half hour before shutting down the machine, in order to prevent rusting the cylinder and valves.

For steam cylinder lubrication a good grade of steam cylinder oil should be used. This oil again to be obtained from a reputable oil company who will give you the proper oil to use for the steam conditions under which you operate. The proper quantity of oil to be fed to steam cylinders is much greater than to air cylinders due to the constant washing away of the oil by the steam. Approximately four times as much oil will be needed in the steam cylinders as in those for air, subject, of course, to variable local conditions. Depending on its viscosity, a pint of steam cylinder oil will furnish from 5000 to 8000 drops.

NEVER USE pipe sizes smaller than pump calls for. For long pipe lines, use two or three sizes larger. Make suction pipe as short as possible and avoid bends and elbows. Make sure that there are no air pockets in the line and that the suction pipe is absolutely air tight. Place a check valve and a gate valve in the discharge pipe, as close as possible to the pump. The check valve must be placed between the gate valve and the pump in order to be able to inspect or repair the check valve without being forced to empty the discharge line. Always provide a strainer on the end of the suction line. This protects the pump from being choked by foreign matter, and insures safe operation.



Why Not More CO_2 Refrigerating Plants?

I HAVE often wondered just why there are so few CO_2 refrigerating plants in this country. It seems to me that the principal reasons are that operators are either ignorant of the system or afraid of it. As far as I can see, the CO_2 machines are as simple to operate as any of the ammonia machines and are in some ways safer.

With ammonia, we have low pressures, while with the CO_2 system we use high pressures. The ammonia plant usually carries a back pressure of 15 to 25 lb. and a high side of about 150 lb. In the CO_2 machine, the pressure on the low side is about 300 lb. and, on the high side 900 to 1000 lb.

When there is an ammonia leak, we are warned by the odor, but we are not warned in this way of a leak in the CO_2 system, as CO_2 is odorless. In a great many places a leak from the ammonia system is seriously objectionable as in hotels, hospitals, department stores and the like.

CO_2 can be used in the preservation of foods; a leak of ammonia in a room full of food will spoil it. Machinery for use with CO_2 is small compared with ammonia machinery of the same tonnage. It is not inflammable and not explosive, therefore, it is safer to store and handle. We can use brass or copper pipe with CO_2, while with ammonia we must use steel or close grained iron, showing a great difference in weight. CO_2 is non-poisonous, as it is used for the carbonizing of soda water that we drink at the soda fountain.

I have talked with operating engineers in plants where the CO_2 and the ammonia systems were both in use. In such plants the men all prefer the former, as they say it gives the least trouble and requires the least attention.

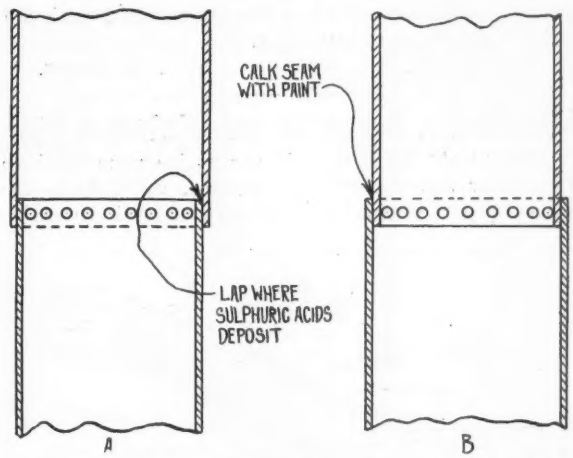
Rochester, N. Y. R. G. SUMMERS.

Reversal of Laps Prevents Stack Deposit

D. K. FRENCH, in the December 15 issue, answers C. M. A.'s inquiry regarding the rusting of a steel stack, explaining that the substance causing-the rust was a product of combustion. He recommends coating the seam and inside of the stack when new with a chemical preparation made especially for that purpose. This, he contends, probably would have prevented the corrosion and considerably increased the life of the stack; but how much, he thinks would be difficult to say.

This chemical preparation should have the desired effect; but when ordering a new stack, if C. M. A. will reverse the lap as shown at B on the sketch, or in other words, turn the stack upside down if all the plates are of the same thickness, he will find that the life of the stack will be more than doubled.

Such a reversal of the laps will prevent the acids on the inside of the stack from depositing between the plates at the laps, and the upturned seams on the outside of the stack can be filled with ordinary stack paint to keep rain and moisture out. He will also find that his stack will



A. INSIDE EDGE OF LAP POINTS UP. B. INSIDE EDGE POINTS DOWN, ALLOWING DEPOSIT TO DRIP OFF

always present a neat and clean appearance because the acids or products of combustion can not deposit in the seams and run down the outside of the stack.

Superior, Wis. E. EMMERT.

Keeping the Plant Clean

REGARDING the articles that have appeared in Power Plant Engineering on keeping the plant clean, it seems to me that this is one duty for which the chief engineer is responsible. I think that it is just as essential to keep the plant clean as it is to keep it in good mechanical condition. No plant, however modern the equipment, will remain long in good operating condition with an accumulation of grease and dirt on everything. When trouble arises in such a plant, it takes two or three times as long to do the work, because it is necessary to clean up everything before starting the repair work. I think that keeping the plant clean should be considered part of the operation, for the satisfaction the operators will get out of it, if for nothing else.

I believe there are few plants that are dirty and at the same time efficiently operated. There is more excuse for a clean plant being inefficiently operated than for a dirty plant being efficiently operated, as all of us will have mechanical troubles in the plant.

Most of us look on the insurance inspector as a kind of unnecessary evil, when in reality he is our best friend.

Get his ideas and suggestions and, if possible, have them in force when he comes again. One inspector told me that when he found a clean plant, he knew everything else was in the same shape and he considered it a good risk.

Cleanliness is important, too, from the viewpoint of the management and the stockholders. Of course they expect results but, when they bring their friends and prospective stockholders to the plant, the visitors are first impressed by the sanitary conditions. Their first impression is their lasting one.

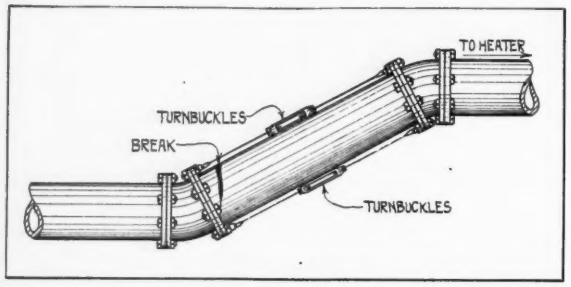
Some engineers seem to think if they had certain new equipment they could keep things in much better condition but if they would just use properly what they have, they would surprise themselves.

I think that, if nothing else will interest the engineer in plant cleanliness, he should think of himself and his family, as there is much less chance of getting hurt in a clean plant than in one where something might catch fire or he might fall over something.

Malvern, Ark. W. H. Howze.

Mending a Break in the Exhaust Pipe

One night the receiver in our engine room filled up with water and an explosion resulted, which broke the exhaust pipe as shown in the sketch and filled the engine



SKETCH SHOWING METHOD OF HOLDING CRACK TOGETHER WITH TURNBUCKLES

room with steam so that we had to shut down. This exhaust line was of 18-in. pipe and the break extended around the pipe for about 9 in., as shown, the edges being somewhat roughened. I made temporary repairs with copper wire and packing and, although the connection leaked considerably on starting again, we stood it until the end of the week.

Then, after we had shut down, we proceeded to make a permanent repair. We fitted some rods and turnbuckles to the flanges as illustrated. Then we filled the crack with a mixture of red lead and iron filings, put on some clamps and drew the turnbuckles up tight. Since that time, there has not been a sign of a leak.

Utica, N. Y. M. M. Brown.

Why Not More Reinforced Concrete in in the Power Plant?

In the Dec. 1 issue appears a cut showing a turbo-generator unit mounted on a reinforced concrete foundation. I wonder if a good many engineers are not using I beams when properly reinforced concrete would do the work, as in this case? Is there not a feeling that there is a big risk in carrying heavy loads on concrete and does not this skepticism come from a belief that it requires a knowledge of higher mathematics to design properly a beam or slab?

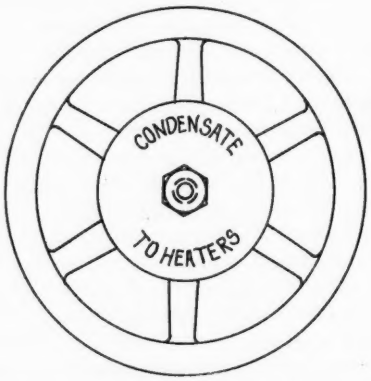
If, as I believe, this feeling develops a tendency to resort to steel beams, I believe that you could well devote some space to explaining in an elementary way how the average engineer can be perfectly safe in his design by simply selecting his beam size, together with required reinforcing, from tables prepared for this very purpose.

The writer has designed a variety of concrete structures, all of which are still intact. We always consider concrete first, as it has all the properties desirable for mill construction. As a reader, I would be very glad to render any assistance possible to the engineer who still clings to the idea that to carry a real load, he must use the old reliable I beam.

Lincoln, N. H. A. P. Nutter.

Indicator Disk Prevents Mistakes in Operating Valves

One method of preventing confusion as to the function of any particular valve in the power plant is to fasten an indicator disk to the valve wheel as shown. The disk can be labeled to show exactly what the valve is for and where the flow from the open valve leads. This disk may be made of brass or other suitable metal and may be fastened to the boss on the wheel by two 8-32 screws.



INDICATOR DISK ATTACHED TO VALVE WHEEL

I have found the device useful not only to new operators coming into the station but also to old employes.

Salt Lake City, Utah. A. L. Donovan.

Keeping Pumps in Good Order

Pumps, like other machines, will give satisfactory service if kept in good order; otherwise, due to wear and development of various defects, they will give trouble. One of the most common of these troubles is lost motion in the valve gear, especially on the duplex type of pump. Wear of this mechanism has always seemed to me to be excessive, even when it is kept well oiled. When the wear is excessive, the stroke of the pistons lengthens and there is frequent stoppage at the end of the stroke. The wear should be taken up by using new pins and bushings. One thing that is sometimes overlooked in repairing these parts is the block-shaped nut on the valve stem that engages with the slide valves. This nut may wear so thin that, even if all the slack is taken out of the rest of the gear, the pump will still long-stroke and bother by stopping at the

end of the stroke. The remedy is to make new blocks slightly thicker than the old ones were when new and square up the faces of the lugs so the blocks will bear evenly and drag the valve straight and square over the seat. Uniplex pump valves get so they have too much play where the lug on the valve engages with the chest piston. The wear may be so much that the valve will move slantwise across the seat.

Piston rings on the steam end will often give trouble, even though, on a superficial examination, they appear to be all right. If they are worn until they are too narrow to fill up the groove in the piston, the steam may pass under the ring or rings and escape to the other side of the piston to such an extent that the pump does not have enough power to do its work. For medium-sized pumps, we prefer two snap rings of reasonably light section set out by a bull ring equal in width to both outside rings. The rings should be doweled so the joints cannot stick together. This construction necessitates a follower plate, as the bull ring cannot be sprung over the piston to get it in the groove.

We have never had it explained to our satisfaction why brass valves on brass seats in our boiler pumps pit so badly, but so far as our experience goes, they do. For this reason, when we get a new boiler pump with brass valves, we usually discard them and install hard rubber valves or valves made of a rubber mixture. There should be at least two sets of valves on hand so that one set can be repaired while the other is in use. We usually have the face turned on the surface grinder. If no grinder is at hand, a stick, with the end turned to fit loosely in the stem hold in the valve, may be used to hold it against the flat side of an emery wheel. If held in the right manner, the valve will revolve and the surface will be ground sufficiently true so that when the heat and pressure come on, it will seat properly. In the absence of the grinder and wheel, the valve may be trued by laying a sheet of emery paper on a flat board or surface plate and rubbing it to a flat surface. With this method, the direction of motion should be changed frequently. We often true up the steam valves of our receiver pumps by laying a sheet of emery cloth on a surface plate and rubbing the valve to a reasonably true surface. Then we lay on the seat a strip of cloth the width of the valve and long enough to reach over the top; the emery cloth is then rubbed back and forth until the seat shows a good surface. This may not be the best method, but if carefully done will give good results.

When a pump refuses for some mysterious reason to do its work, the application of a pressure gage to various parts of the pump and piping will sometimes show interesting facts.

Anderson, Ind. J. O. Benefiel.

Extracting Splinters

When a workman gets a splinter, either of wood or metal, into any part of his body, it may be removed by the following simple method, which is not at all new but is not generally known.

First fill a wide-mouthed bottle nearly full of very hot, clean water. Then, holding the bottle on a slant so that the water will not touch the flesh, press the mouth of it over the splinter. The condensation of the vapor will cause a vacuum, which will draw the flesh down, and it will be found that the splinter will come forward far enough to allow it to be caught with a pair of small pliers and pulled out.

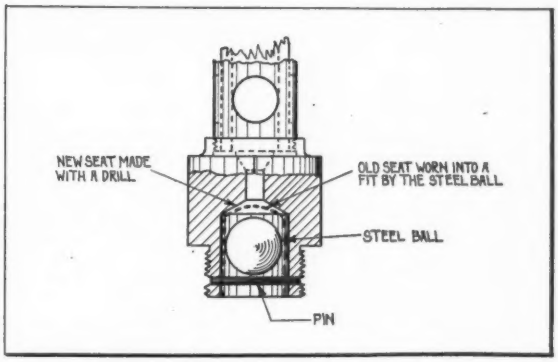
I have seen an iron splinter, the end of which was an eighth of an inch under the skin, extracted quickly in this way.

Toronto, Ont. John Thorn.

Lubricator Check Valve Repairs

After continuous use for a long time, our gas engine cylinder lubricator refused to work, the oil remaining in the sight glass and failing to reach the piston, so that it was necessary to stop the engine.

An attempt was made to remedy the condition by opening the filler plug, running a wire through the small air pipe and washing the lubricator in gasoline, but still the oil would not pass the check valve. Finally, the lubricator was taken apart and it was found that the steel ball forming the check valve had worn its seat into the brass so deep that it took but very little oil to make it stick, thus preventing the explosion pressure from being released and the oil from running down to the piston. This condition is shown by the dotted lines in the sketch.



SKETCH SHOWING HOLE REDRILLED TO GIVE NEW SEAT FOR BALL

In order to make a new seat for the ball, a drill slightly larger than the original hole was run in as shown. Then, on reassembling the lubricator, it worked perfectly.

While taking this lubricator apart, one of the cork washers under the large glass was broken. This washer was replaced by one cut from an old shoe top. We believe that this leather washer works better than the cork one did.

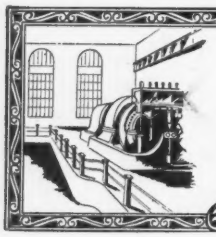
Death Valley, Calif. Chas. Labbe.

Washing Machine Not Approved

Mr. Carother's washing machine, described in the Dec. 1 issue, page 1222, would find little favor in our plant, as we discourage the use of high-priced high pressure steam for such operations. While the device is ingenious, I do not feel that it should be recommended for installation in a plant where steam costs good money.

A. P. Nutter.

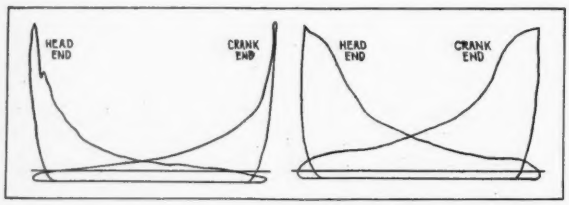
Superpower does not supplant existing systems but supplements them. It may be defined as the saving made by interconnecting generating stations whose peak loads are reached at different times, and so distributing the loads among stations of different unit operating costs as to deliver power at minimum cost.



QUESTIONS AND ANSWERS

Why Does Riding-Cutoff Engine Pound?

OUR RIDING-CUTOFF engine, which gives indicator cards as shown, has always pounded badly, the pounding growing worse when the engine is run condensing. The cylinder has been rebored and now there is a noise in it, especially when higher steam pressures are used. This noise decreases, however, when the pressure is reduced to about 110 lb. E. H. W.



INDICATOR CARDS FROM 18 BY 27-IN. RIDING-CUTOFF ENGINE. SPEED, 120 R.P.M.; STEAM PRESSURE, 130 LB.

Allowing for Expansion in Underground Pipe Line

WE ARE planning to run a line of 350 ft. of 5-in. pipe in a straight run underground to carry 225 lb. pressure with 100 deg. superheat. What would be the maximum distance one expansion joint should be expected to take care of? Would bends be preferable to slip joints for this work? A. G. S.

A. In regard to the expansion of 350 ft. of 5-in. pipe in a straight run under the conditions of pressure and superheat mentioned, the method you use to provide for the expansion seems to us to depend on the amount of space that you have available. The slip joint, of course, does not occupy as much space as the bend. A slip joint for 5-in. pipe will take care of 4 in. of expansion, according to the tables of a well-known manufacturer. The total expansion of 350 ft. of pipe, which will have a total temperature of 497.4 deg. when steam is flowing, will be 15.72 in. (Expansion due to change in pressure is small here.) The expansion is figured from the formula:

$$L_t = K \times (T_1 - T \times L)$$

in which L_t is expansion in inches and K is the co-efficient of expansion of wrought iron pipe, or 0.000,008,95. T_1 is the temperature of the steam, T the initial temperature, assumed to be 70 deg. and L the length of the pipe in inches. To take care of this with slip joints, four such joints would be necessary. If these joints can be installed in such a way that they can be easily inspected and repaired, they should give satisfactory service.

Many engineers prefer to use bends wherever possible. The disadvantage of using bends for this particular line is that the bends must be drained on either side and for underground work, such as you propose, this might prove an inconvenience. In addition to that, you might not have room to install two or three large bends to take care of the expansion and might have to use several small ones, thus complicating the drainage problem. The amount of expansion that can be taken care of by a bend depends on its radius. For example, with a radius of 60 in., or 5 ft., a circle or double-U bend of 5-in. pipe would take care of 8¼ in. of expansion.

Ammonia Compressor Capacity Depends on Pressures

WHAT TONNAGE will be produced by a 4 by 4-in. ammonia compressor, driven by a 5-hp. motor at 225 r.p.m., using 16-lb. suction pressure and 185-lb. condenser pressure? I have been told that this machine will do 4 T. of refrigeration with a 5-hp. motor, but do not see how this can be done without overloading the motor. Why is it that the size of motor varies so much with different makes of small machines of the same tonnage? R. S.

A. To find the refrigerating capacity of a compressor at any given pressure, multiply the area of the piston in sq. in. by its stroke in in. and the number of strokes per minute and divide by the number of cu. in. given by the accompanying table, which shows the relation between capacity of the cylinder temperature of the suction gas and suction pressure. In this case, we have used 7500 corresponding to a suction pressure of 15 lb. gage. Then, substituting in the above formula the values you have given, the formula becomes—

TABLE OF CYLINDER CAPACITIES FOR VARIOUS PRESSURES

Lowest Temp. to be Maintained. Deg. F.	Approx. Temp. of Suction Gas. Deg. F.	Suction Pressure Gage Lb.	Cu. In. of Cylinder Per Minute.
— 5	—17	5	10,700
0	— 8	10	9,000
+10	0	15	7,500
+15	+ 6	20	6,000
+20	+11	25	5,400
+25	+17	30	4,800

$$(4 \times 3.14 \times 4 \times 2 \times 225) \div 7500$$

which equals 3.01 T. per 24 hr. We have assumed here that the compressor is double acting.

There seems to be some disagreement among the various manufacturers of refrigerating machinery in regard to the basis of rating it. It is customary, in most cases, to rate such machinery on the basis of the abstraction of an amount of heat equal to the heat of fusion of one T. of ice per 24 hr. This amount is approximately 288,000

B.t.u. It is also customary to specify the temperatures and pressures between which this abstraction of heat takes place. In this country, it is customary to rate the machine at 16 lb. gage suction pressure corresponding to a temperature of 0 deg. F. and 185 lb. gage condenser pressure corresponding to a temperature of 95.5 deg. F. On this basis, a table given in Marks Mechanical Engineers' Handbook has been worked out for the theoretical hp. to produce one T. of refrigeration. This amount, corresponding to the pressures you have given, is 1.147 hp. Adding 50 per cent to this for probable actual horsepower, as noted in the table, we have 1.72 hp. per T. of refrigeration necessary. If we divide 5, the horsepower of the motor driving the compressor, by 1.72, the result is 2.9 T. of refrigeration.

Another table given in Kent's Mechanical Engineers' Handbook gives the same result. The tonnage computed from these tables is close to that computed by using the formula in paragraph 1, close enough to be considered a check. We believe, therefore, that, under such conditions, your compressor should be rated at 3 T. It should be noted, however, that a decrease in the condenser pressure would increase the rating of the machine.

Compressors used for ice-making purposes must be of from 90 to 100 per cent greater capacity than those used for refrigeration, due to the fact that, in ice-making, the water must be cooled from the temperature at which it leaves the cooling coil, say 85 deg. F., to the temperature of the freezing tank, which may be 14 deg. F., and in addition to this extract 143.5 B.t.u. of latent heat for each pound of ice.

Exact size of compression cylinders for any given refrigerating capacity cannot be given unless the temperature to be maintained in the cooler is known. For general purposes it is safe to say that a compressor having 7500 cu. in. capacity per min. will handle sufficient gas for one T. of refrigeration during a continuous run of 24 hr. This rule is based on an expansion pressure of 15-lb. gage, with cooling water at 70 deg. F. If the pressure in the expansion coils varies from 15 lb. the number of cu. in. of cylinder capacity required for one T. of refrigeration will also vary.

Economy of Increasing Engine Vacuum

WILL IT be more economical, in operating a tandem-compound Corliss engine, using a condenser but no feed-water heater, to operate with 22 in. of vacuum in order to get hotter feed for the boilers or to operate with 26 in. and let the boiler do more of the work? E. H.

A. The question of whether it is more economical to run your tandem compound engine with a vacuum of 22 in. or one of 26 in. is difficult to answer, as you have not given enough information for a proper mathematical calculation. In general, however, there is an advantage in increasing the vacuum on an engine up to 26 in. The theoretical gain by doing so in your particular case would probably be about 8 per cent in the efficiency of the engine. This is figured on the basis of the whole engine referred to the low pressure cylinder, which, under present practice, might be assumed as 25 lb. per sq. in. The decrease in back pressure converted from inches of mercury to lb. per sq. in. would be 1.98 lb. so that the gain would be

$$(100 \times 1.98) \div 25 = 7.92 \text{ per cent.}$$

This advantage, however, is more apparent than real, because it is probable that the power used by the pumps and other auxiliary apparatus in handling the increased supply of condenser water necessary at the higher vacuum would decrease that amount by as much as 4 or 5 per cent. For example, the amount of condenser water in lbs. necessary to condense a pound of steam under your conditions equals

$$H_1 - H_2 - \frac{2546}{W} - (T_1 - 32) \div (T_2 - T_0)$$

In this formula H is the heat of 1 lb. of steam at the initial pressure, assumed 135 lb. ga; H_2 is loss by radiation, assumed .01 of H_1; T_1 is the temperature of the condensate; T_2 is the temperature of the discharged cooling water; T_0 is the initial temperature of the cooling water, which can be assumed in this case to be 70 deg.; W is the water rate, assumed to be 14.

When using a 22 in. vacuum the numerator is found from the steam tables to be 893.6 b.t.u. Then since the temperature of the discharge water T_2 will be on the average 20 deg. less than the corresponding theoretical temperature in the condenser for the 22 in. vacuum, T_2 will be 132 deg. Substituting these values in the above formula, we find that with a 22 in. vacuum 13.3 lb. of water are necessary to condense 1 lb. of steam.

Considering the conditions with a 26 in. vacuum, the theoretical condenser temperature will be 125 deg. and the probable temperature of the discharge water will be 105 deg. Using the same formula as before and finding from the steam tables the value of the numerator to be 911 b.t.u., by substituting these values in the formula, we find that 26.1 lb. of water must be used per lb. of steam condensed. It can be seen that much more condenser water must be used and it seems probable that the power necessary to handle this additional water would be a considerable amount. Considering the advantage of using hotter boiler feed water, you would gain about 1 per cent for each 10 deg. rise in temperature of the water put into the boiler. That is, if the feed water temperature with the 26-in. vacuum is 105 deg. and the temperature with the 22-in. vacuum is 132 deg., the net saving will be about 2½ per cent with the 22 in. vacuum.

Boilers Should Not Be Drained Under Pressure

IN THE plant, of which I have just taken charge, I find that the firemen, when draining a boiler, run the pressure up to about 90 lb. to have steam enough to blow tubes and then with about 30 lb. pressure, they blow all the water out of the boiler. Is there any advantage in doing this? H. E.

A. We do not believe that it is desirable for your firemen to drain a boiler while it is under pressure. In order to blow tubes they will need considerable steam and by running pressure up to 90 lb., they ought to have enough. Then they should allow boiler and setting to cool before draining the boiler. When the water is blown from the boiler under pressure, the metal surfaces and the brick work are still extremely hot. This heat has a tendency to bake the scale and sediment onto the steel and makes it, in many cases, difficult to remove, so that nothing is gained by this method. On the other hand, by allowing the boiler and setting to cool gradually, much of the sediment remains soft and can be washed out; furthermore, there is not so much danger of the setting being strained by too rapid a change of temperature.

EDITORIAL COMMENT

Fuel Problem of Great Interest

If anybody questions either the interest with which engineers of today are delving into the fuel problem or the advancement which has recently been made in the practice of generating power from coal, let him refer to the two regional engineering meetings which were recently held in Chicago and Cleveland.

Economic importance of coal production does not cease to exert its influence when the coal is delivered at the mine mouth. The engineering profession must assume, as its responsibility at that point, the problem of transportation and the proper regional distribution in order that the greatest possible portion of the coal mined may be delivered to the power producing plant upon an economical basis. At that point the problem must again be taken up so that this coal will be burned in such a manner as to add still more to the economic value of our resources.

Upon this latter phase of the fuel problem were concentrated these two regional meetings. At Cleveland, some of the best engineering talent in America devoted its attention to the problem of burning fuel in the pulverized form. One hundred years of experimentation have been devoted to this problem in order that today we may obtain a kilowatt-hour output for an expenditure of 17,300 B.t.u. in a furnace. The persistence with which the engineering profession has worked at that problem resulted last year in the successful use of 20,000,000 T. of pulverized coal.

At Chicago the problem of the economical production of power was attacked from the viewpoint of combustion control, air preheating, furnace construction and prime movers. This meeting was marked with the same degree of enthusiasm as that which was displayed at Cleveland, thus indicating that the engineer knows what his job is and how he should go about it to get results. He knows that what he has done today must be bettered tomorrow and he has set out with determination and enthusiasm to do the job right.

Source of Power a Question of Economy

Many engineers in industrial power plants have viewed with alarm the growing use of central station power in factories and the replacement of isolated power plants with equipment driven by electricity generated in public utility plants. Human nature seems to resent rather than invite competition and the question of the central station vs. the isolated plant is purely one of business competition as to which will be the more economical for the owner of the plant. The inefficient plant must go, likewise the engineer who cannot meet competition must seek work at which he can make a profit for his employer. There is, however, no reason to believe that the central station will replace a more economical industrial plant.

Motor driving of factory equipment began to come out of the experimental stage about 25 yr. ago and it is interesting to note that up to the present time there has been a steady increase in the capacity of motors installed in factories generating their own power. Recent statistics show these figures to be 7,000,000 hp. and the curve is almost a straight line from 1,000,000 hp. in 1904. In factories driven by central stations, however, the motor capacity has advanced much more rapidly during the past 10 yr. until today they have 12,000,000 hp. of motors.

These statistics show that the factory power plant is maintaining its former rate of increase but that the central station is increasing much more rapidly. Small factories located in loft buildings and rented quarters and others that do not use exhaust steam in their processes have taken advantage of the improved service which central stations can now render and have ceased to operate their small, uneconomical generating units. Add to these changes the growth in small manufacturing establishments and we have, in general, the sources contributing to the rapid growth of central station power in factories. The efficient factory plant is still operating and many more are being built throughout the country.

Specialists and the General Practitioner

General practitioners in medicine may be, and frequently are, good diagnosticians, but it frequently happens that they have not the inclination to attempt specialization. The standards of practice of the whole medical profession, however, are constantly being raised by the development of new methods and improved equipment by men who concentrate all their time and energy on only one phase of the work.

In many respects, the work of the specialist in medicine in developing improved methods and equipment that can be used by the general practitioner is analogous to the work done by the large central station and its relation to that of the average industrial plant. Engineers often comment on the fact that it is usually in the former that the important research work is done leading to simplification and improvement in existing methods of power generation and to the working out of new designs to cope with constantly changing necessities.

On reflection, it can be seen why this is so. The business of the central station is to produce one thing only; electric power. The total cost of its product, therefore, is the cost of power. What more natural than that it should bend every effort to reduce that power cost?

Yet the smaller industrial plant with its variety of problems, of which the production of power is only one, can often use to good advantage the methods and equipment first developed by the central station just as the general practitioner in medicine uses the things developed by the specialist. An interesting illustration of this occurred at the recent meeting of the Chicago section of the A. S. M. E. Following a discussion of a paper on water-cooled furnace walls, which presented the subject mostly

from the standpoint of the large central station, the announcement was made that in a certain industrial plant two new boilers are being installed, one with the ordinary brick refractory furnace, the other with a plain-tube water-cooled furnace. After running the two boilers under identical conditions for several months, keeping careful operating records, the brick furnace will be replaced by a fin-tube water-cooled furnace and further observations made. Valuable data should be obtained in this plant.

It is not to be expected that the smaller industrial plants can spend the time and money for experimental purposes. Such plants are working in a much different field. But, since both types have in common the all-important problem of lowering power costs, it is desirable for the industrial plant to adapt to its own conditions the improvements that are applied successfully by specialists in the field of power generation.

Only Engineering Can Secure Modern Efficiency

Among the factors in power plant operation which contribute to inefficiency is the over confidence which many owners have in the ability of the equipment which they have purchased to operate without human supervision. Great strides have been made by manufacturers in producing machinery that requires little attention and quite a number of automatic substations and even hydroelectric generating stations are now in successful operation. Such equipment is unquestionably replacing many men of the laboring type whose work is almost wholly mechanical but machinery cannot be made to reason and must, therefore, have supervision.

In plants equipped with mechanical coal and ash handling machinery, automatic stokers, combustion control devices and the like, common labor has almost disappeared but unfortunately too many owners have retained common laboring men to operate this equipment with the result that their plants are neither as efficient nor as reliable as they should be.

Regardless of how far we may go towards making machinery automatic, it must still be regarded as a tool in the hands of man. Heavy manual labor is being eliminated but in its place must come skill in the application of scientific laws to the changing conditions about the plant. This is work for a skilled engineer, capable of recognizing dangerous or unwanted conditions and of keeping equipment under his care in first-class condition.

The Evolution of Superpower

No branch of engineering has developed more rapidly than that devoted to the generation and application of electricity. Although less than 50 yr. old, the electrical industry has grown to be one of the most fundamental of all industries and in any country, the extent of its development is largely a measure of civilization in that country. This, of course, is true of any form of power, but as will be shown, it is particularly true of electricity.

In order to realize the full significance of electrical power as a factor in civilization, let us consider briefly the history of power. To do this, we must go back to the stone age when the only form of power man was able to apply to his use was his own muscular power. True enough, all sources of power we use today were in existence then, but man had not learned how to apply the power to his needs.

Later on, man learned to make use of the wind by compelling it to propel his boat. This at once made things easier for him as it enabled him to transport himself and his belongings from one place to another with far less effort than had previously been necessary.

Still later, he realized the force of falling water and conceived the idea of the water wheel. These were crude affairs at first but they placed far more power at his disposal than he had ever had before.

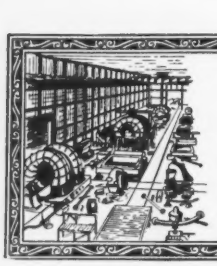
Man's real release from the bondage of muscular effort came with the invention of the steam engine. Here was a device by which one or two men were able to do the work of hundreds. It multiplied man's efforts a thousand times. Furthermore, this device would keep on working 24 hr. a day, 7 days a week. It did not get tired. As a consequence, man had much more leisure in which to think up new ways of utilizing the power produced by the engine.

Notwithstanding the new degree of freedom afforded man by the coming of the steam engine power was still a local affair. It could only be utilized at or within a few hundred feet of the place where it was produced. Communities grew up, therefore, in the immediate vicinity of the sources of power. In New England, towns and villages appeared along the power producing streams and rivers. In other sections of the country they grew up around the steam plants which were located where fuel was available.

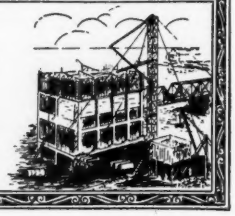
Then came electricity, and once again civilization took a jump forward—an enormous jump this time. With the development of electricity supply systems it was no longer necessary that power be used at the point where it was generated. For the first time in history, large quantities of power could be produced in one place and used in another. The power of water falls in remote places could be made available for use in industrial communities many miles distant. Electricity made an abundant supply of power available anywhere, so to speak.

What was the result? You see it today in the thousands of flourishing industrial and social communities that stretch from one end of the country to the other. You see it in the huge transmission lines that deliver hundreds of thousands of kilowatts of electrical energy from the remote mountain rivers to the great cities of today. You see it in the electric light which you snap on in your home; you see it in your radio, in your automobile, in your telephone. There is practically nothing in your life today in which electricity has not at one stage or another played a part.

Today we hear a great deal of superpower. What is superpower? We discuss it as though it were something new. But it is not new. The history of superpower is the history of power itself and as we have shown, dates back to the stone age. We have referred to the manner in which electrical distribution systems made an abundant supply of power available at a large number of different points. Such systems are to be found in thousands of cities and towns throughout the country. Instead of, as formerly, each user of power having his own generating plant, today we have hundreds and thousands of users drawing power from one system. Such systems were the first superpower systems—small ones, it is true, but superpower systems nevertheless. Today we are engaged in a process of connecting these separate distribution systems together by means of tie lines and it has become customary to refer to such connected systems as a whole as superpower systems. Superpower of today, therefore, is not a new development but merely another stage in the natural economical development of one great network of electrical distribution systems which will some day serve every city, town and hamlet throughout the nation.



POWER PLANT ENGINEERING DEVELOPMENTS AND NEWS



Annual Power Meeting of Chicago A. S. M. E.

PROMINENT ENGINEERS AT TWO-DAY MEETING DISCUSS LARGE UNITS, FURNACE REFINEMENTS AND OTHER WAYS OF RAISING EFFICIENCY. MEMBERS VISIT CRAWFORD AVENUE

DISCUSSING the most recent and interesting developments in the modern power plant, the speakers at the second Power meeting of the Chicago section of the A. S. M. E., held on January 14 and 15, 1925, at the LaSalle Hotel, Chicago, told how water-cooled furnaces, 50,000-kw. turbines, air preheaters and new methods of auxiliary drive have come to the front in the past few years.

The annual banquet of the society was held on the evening of Wednesday, January 14, at the LaSalle Hotel, the speakers being David R. Forgan, who discussed "War and Credit," and Frank D. Chase, who took up "The Modern Industrial Plant and Its Power." As a fitting climax to the meeting, an inspection trip to Crawford Avenue was made on Thursday afternoon.

"Combustion Control" by T. A. Peebles of the Hagan Corp., was the first paper and was an interesting presentation, first of what combustion control equipment is not, and second of what it is. Such equipment, the speaker declared, is not intended to replace the fireman, but to give him more opportunity to maintain proper conditions in his fire, by automatically regulating the ratio of air and coal supplied for various loads and by regulating the quantity of air and coal.

While it is not difficult to adjust the ratios and the quantity of air for a given load, the quantity of coal cannot be varied so easily. In any event, the fireman himself, in making adjustments to take care of variations in load, will overadjust in almost every case. This overadjustment can be prevented, especially if the load is irregular, by well applied control apparatus.

The margin between the average efficiency of operation and the maximum efficiency obtained on a test is only 3 or 4 per cent in a large central station; this margin is from 4 to 10 per cent in a good industrial plant; while in the average small industrial plant, it is from 15 to 50 per cent. Properly designed and well applied combustion control equipment will reduce these margins by at least one-half, the speaker said.

WATER-COOLED FURNACE WALLS

In the second paper of the first morning session, "Water-Cooled Furnace Walls," H. D. Savage of Combustion Engineering Corp., discussed the more recent developments in the construction and operation of the finned-tube type of water-cooled walls. He stated that this type of furnace now appears to be an article of reliability and commercial merit, because for over a year three such furnaces have been used with stokers at Hell Gate and six with pulverized fuel at Sherman Creek with no unusual operating difficulties.

At Hell Gate, the boilers have been run at ratings of about 600 per cent, tests showing as high efficiencies as have been obtained anywhere and indicating that the prophecies of trouble due to burning of fins, erosion of tubes and excessive furnace cooling, have not been fulfilled. Welding of fins to tubes insures good heat transfer and it has been found that cost of installation is not greatly in excess of that of the latest types of refractory furnaces. Furthermore it has been found that furnace maintenance charges are greatly reduced by the use of the fin furnace.

Possibilities of increasing ratings of existing boilers by use of water-cooled furnaces, instead of adding more boilers to increase the capacity, are also apparent. Further refinements in the design of fin furnaces will no doubt be made, said Mr. Savage, and the subject was presented here not as a technical paper, but with the hope that the discussion of it would arouse interest in this successful and worthwhile engineering development.

ACTION ON PIPING CODE

At the close of the morning session, a brief discussion of the proposed Piping Code, read by Mr. Doll of Crane Co., led to a vote by the meeting that the executive committee president of the Chicago section appoint a committee of suitable size to consider the subject in conjunction with the New York section.

In the afternoon session of Jan. 14, Robert Cramer, consulting engineer of Milwaukee, dealt with the subject, "Uniflow vs. Compound Duoflow Engines." In relating the history of the development of steam engines, he stated that early design was based on indicator diagrams rather than thermodynamic principles, thus practical design preceded theoretical considerations.

Among the important losses in steam engines is the initial or cylinder condensation. In the duoflow engine this may be reduced by compounding. The uniflow engine is an attempt to do in one cylinder what the duoflow engine does in two or more cylinders. By providing the exhaust outlet in the middle of the cylinder which is at the end of the piston stroke, the steam does not sweep the cylinder surfaces; the variation in the temperature of the cylinder walls is not so great as it is with a duoflow engine so that initial condensation is reduced. The temperature of the steam remaining in the cylinder is raised, during the compression stroke, to a point considerably above the exhaust steam temperature and with the true uniflow engine, the compression pressure is high, which necessitates a larger cylinder than would be required with

the duoflow engine and due also to the necessary over travel of the piston past the ports, expansion in the uniflow engine is not as complete as in the duoflow.

Uniflow engines must also have large clearance in order to reduce the compression pressure. Modifications of the true uniflow type are provided with auxiliary exhaust valves which permit later compression and are a compromise between uniflow and duoflow engines.

Uniflow engines give a flat economy curve when compared with a duoflow engine designed for similar conditions. The duoflow compound or triple-expansion engine has one best economy load and if properly designed can give better economy for specific conditions than a uniflow engine can give. Where the load varies, however, the overall economy of the uniflow engine is better than that of the duoflow. For this reason industrial plants are installing more uniflow engines today than other types. This in the opinion of Mr. Cramer is the recognized field for the uniflow engine. Where the load is steady, however, the plant is best served by a compound or triple expansion duoflow engine.

"Large Steam Turbines" was the subject of a paper read by Francis Hodgkinson, chief engineer of the Westinghouse Electric & Mfg. Co. This paper related the progress in steam turbine design, stating that the most difficult problem in the early designs was to secure economy in the high pressure range. At the present time, however, designers are turning their attention to the low pressure range; that is to the manner of getting the exhaust steam away from the turbine blades before its energy has been completely exhausted and the turbine must force it out into the condenser.

Mr. Hodgkinson emphasized the fact that he considers reliability as the first thing to be considered in turbine design with gain in efficiency as the second consideration. He touched on the advantages of stage bleeding and stated that higher steam pressures give possible improvements. Reheaters necessarily complicate steam piping and regulation and designers are trying to reduce these complications or eliminate reheaters entirely. He is of the opinion that designers will not hold back on steam pressures commercially available, that they can design their turbines to withstand pressures which are likely to be employed in practice.

"The Diesel Engine" by L. H. Morrison

As a result of the phenomenal growth of industry and its crowding into cities during recent years, with its resultant advantages in financial control and disadvantages in handling of personnel, congestion of transportation and inflation of living costs, there has been a trend in some instances toward decentralization, especially in those industries in which the cost of power is only a small proportion of the manufacturing cost. This decentralization has been effected in the shoe-making industry, the automobile industry and others. It is true that where an industry requires a comparatively large amount of power per unit of product, the small factory plan cannot be adopted. But in such small factories, as well as in farming, a study of the Diesel engine as a source of power is worth while, Mr. Morrison said.

With a fuel consumption per b.hp. practically the same in all sizes and an overall efficiency usually higher than a steam plant, according to the speaker, the Diesel engine is frequently the best small prime mover.

Even when considerable steam and hot water is needed for process work and heating, it may be possible to utilize the heat in the exhaust and in the jacket water for this purpose.

Diesel engines are now in service built in sizes up to 3000 hp. and in Europe sizes of 6500, and in a variety of types—two-stroke-cycle, four-stroke-cycle, solid fuel-injection, air injection and others. The costs of operation of the Diesel can be estimated in advance, and in general these costs, including repairs, lubrication and the like, are low. Although the Diesel engine cannot compete with large steam units for central station work, the speaker said, it has a legitimate field, and in the study of industrial power plants, it should not be left out of account.

Preheating Furnace Air

In covering briefly the subject of "Preheating Furnace Air," on the morning of the second day, C. W. E. Clark of Dwight P. Robinson & Co. gave first a short account of the historical development of air preheaters, mentioned the three types of such apparatus—namely, plate heaters, tubular heaters, and rotative heaters—and then discussed some of the advantages of the plate type which is used in most preheater installations.

With this type of heater, said Mr. Clark, there is generally little trouble from dirt. An improvement in combustion conditions of course results, although in some cases erosion of furnace refractories is caused, especially with pulverized fuel. This can be prevented by using water-cooled walls.

Mr. Clark stated that in his judgment the value of the use of preheated air is high, especially in the central station, and in combustion with stage feed water heating. In one case noted, the heat rate, without considering combustion efficiency or other heat losses, was 12,165 B.t.u. per kw.-hr. using steam air heaters and economizers, while using flue gas preheaters and four-stage bleeding, this heat rate was reduced to 12,085 B.t.u. per kw.-hr. Before a definite answer can be given as to the value of air preheating, the speaker remarked, more data must be obtained.

G. G. Bell of the West Penn. Power Co., in the last paper of the second session, on "Boiler and Turbine Room Auxiliaries," discussed the relative merits of the four general methods of driving auxiliaries. These are (1) the use of current from the main bus for driving motors, (2) the use of a separate a.c. turbine-driven house generator, (3) the use of a spinning house generator and (4) the use of an auxiliary generator mounted on the shaft of the main unit. The latter, he said, is on the whole as efficient and more reliable than any of the other methods, and is used in plants of his company.

Suitable bus connections are made to tie these auxiliary generators to the main circuits if necessary. On circulating pump and fan drives, duplex motor drives are used, one motor to carry normal loads, and a larger one, thrown in by a magnetic clutch to handle overloads, allowing the rotor of the smaller motor to idle.

In general, the speaker stated, very little trouble has been experienced with this method of driving auxiliaries and it is believed to be the most satisfactory one.

The inspection visit to the Crawford Avenue Generating Station of the Commonwealth Edison Co. on Thursday afternoon was a feature of the meeting that was enjoyed by all who attended. Crawford Avenue Station is perhaps the most talked of station in the world and this opportunity to visit it was considered a real treat.

Although there were many things of interest at the plant, the things of greatest interest to the visitors were the three turbo-generator units which constitute the initial installation. These are of three different makes, one a Westinghouse unit, one a General Electric and the third, a Parsons unit made in England. The three units are of approximately the same capacity, i. e., 50,000 kw. At the present time the General Electric machine is the only one in operation, the other two units being in various stages of erection.

Each generating unit is supplied by a battery of five boilers, four of which are regular steaming units generating steam at 500 lb. pressure, while the fifth is a reheating boiler, used to reheat the steam after it has passed through the high pressure stages of the turbine and before it is delivered into the low pressure stages.

The visitors were taken to and from the station by means of motor busses and were conducted to the various points of interest by a corps of competent guides, through the courtesy of the Commonwealth Edison Co.

Pulverized Fuel Discussed at Cleveland

In One-Day Conference, Prominent Combustion Engineers Deal with Problems Encountered in Application of Pulverized Coal to Power Plants

PREPARATION of pulverized fuel and the operation of plants making use of fuel in this form were the principal subjects at a mid-west conference on steam production economies held Jan. 13 at the Hotel Winton, Cleveland, O. This conference, which consisted of three sessions, was held under the auspices of the Cleveland Engineering Society in co-operation with the Cleveland section of the A. S. M. E. and the Cleveland Chapter of the American Society of Heating and Ventilating Engineers.

District Heating

At the morning session, Walter Kline of the American District Steam Co. presented a paper on the subject of group plan heating for apartment and residential districts. This paper told of the experiments of Holly in 1887 which were the pioneer work in this phase of engineering. Mr. Kline pointed out that the advantages of district heating were, fuel saving, convenience, cleanliness and the elimination of street traffic due to fuel delivery and ash removal. This paper was illustrated to show the work incidental to the laying and construction of underground steam lines and to emphasize the importance of good pipe insulation and proper drainage of the soil near pipe lines. It was stated that in Detroit some work has been done towards the elimination of condensation by using dry saturated steam and delivering it through pipe lines at velocities up to 70,000 ft. per min., thus taking advantage of the pressure drop to gain a small percent of superheat at the delivery end.

Preparation of Pulverized Fuel

Treatment of the subject of pulverized fuel was in two divisions, the first being on the subject of fuel preparation and the second on the operation of plants burning pulverized fuel.

History of the development of pulverized coal as a boiler fuel was the subject of the first paper of the afternoon session which was given over to the preparation of pulverized fuel. This historical sketch was presented by H. W. Brooks, fuel engineer, Bureau of Mines, Pittsburgh. It was interesting to learn from that paper that this "new" method of preparing coal has actually been experimented upon since 1818 and therefore our practice of today is based upon over 100 yr. of research and experience. The first applications of pulverized coal failed because of improper furnace design and inadequate and poorly designed pulverizing equipment.

Much credit is due the cement industry for the development of the use of this form of fuel preparation and use. By 1916 the complete success of pulverized fuel was assured in the industries and by 1920 central stations were using it successfully. At the present time over 20,000,000 T. of coal are being pulverized each year.

Central pulverizing plants and their operations was the subject treated by Henry Kreisinger, research engineer, Combustion Engineering Corp., and H. G. Barnhurst, Advisory Engineer, Fuller-Lehigh Co. Mr. Kreisinger stated that dust problem in a preparation plant always gives some trouble and that it could not be solved by wetting down because any moisture added had to be removed later or trouble would result in pulverizing, transporting and burning the fuel.

He stated further that it was best to crush the coal to ½ to ¾ in. because it was then more easily dried and also had the accumulative effect of increasing the capacity of the pulverizer mill. Ordinarily eastern coals need not be dried; there are, however, exceptions to this. Illinois coals and the lignites of the west can be best used when dried although Colorado lignites, containing 22 per cent of moisture, have been pulverized and burned without drying. Waste gas or steam driers are generally preferred to the rotary type because they take up less space, require less attention and give satisfactory results at low costs of operation.

About 1/3 of the flue gas is drawn through the waste gas drier and low pressure steam from auxiliaries can be used in the steam drier, thus in the latter case the drier assumes the role of a condenser and a portion of the heat is reclaimed by the increased temperature of the coal. The coal is ordinarily not heated over 150 deg. F. at which temperature there is no wastage due to the distillation of the lighter volatiles. The whole question of temperatures, however, is dependent upon the dew point of the air or gas, in other words, the temperature must be kept well above the dew point to avoid moisture being added to the coal by the air or gas.

Storage bins may be of either steel or concrete and they need not be insulated unless they are subjected to sudden changes of temperature. It is advisable to ventilate all bins with a small amount of warm dry air. The fire hazard in bins is practically negligible and since the fire always occurs on the surface it can be extinguished by excluding all air from the bin.

Furnaces, built of refractory materials, should be proportioned upon the basis of 20,000 B.t.u. per cu. ft. per hour, or, if this rate must be exceeded, it is advisable to install water-cooled surfaces.

Drying coal is often a seasonable problem according to H. G. Barnhurst, who stated further that any standard

type of crusher is suitable for the initial preparation of pulverized coal and that both the lifting and pulley type of magnetic separators had proved successful. He has found that the vertical burner is best adapted for use with low volatile coals because the heat rising in the furnace greatly improves the flame conditions.

Unit pulverizing plants and their operation were described by W. H. Pettibone, chief engineer, Aero Pulverizer Co. He stated that with the Aero pulverizer, 75 per cent of the air necessary for combustion was admitted to the furnace with the coal, thus giving a shorter flame and so requiring a less depth of furnace and correspondingly smaller furnace volume.

This same subject was treated somewhat from the viewpoint of European practice in a paper which was read for Gould Contant, chief engineer for the Furnace Engineering Co.

Operating with Pulverized Fuel

After an informal dinner, the evening session was opened by E. G. Bailey, chairman of the Cleveland Section of the A. S. M. E. This session covered the operation of the plant and it brought out some valuable information from engineers who are thoroughly conversant with the burning of pulverized fuel, more particularly in large units. The tests, recently conducted at the Lake Shore station, Cleveland, were outlined by John Wolff, mechanical engineer, Cleveland Electric Illuminating Co. They were run on duplex Stirling boilers, rated at 30,600 sq. ft. and equipped with both superheaters and economizers. The water screen at the bottom of each furnace contained 835 sq. ft. and the furnace volume above the screen was 26,000 cu. ft. A feature of this installation is that changes in coal are automatically compensated for in the control system. The individual boiler control is based on the total air flowing through each boiler and each control is set to maintain a given CO_2. About 26 T. of coal is required to bring one of these boilers from room temperature up to the temperature corresponding to 100 to 150 per cent of rating. The plant records show that the net operating efficiency for this installation was 88.2 per cent for the month of November, 1924.

Description of the Trenton Channel Plant of the Detroit Edison Co. was presented by P. W. Thompson, chief assistant engineer of power plants for that company. At that plant the fuel preparation equipment is housed in a separate building and the breechings are equipped with Cottrell precipitators, both of which aid greatly in the problem of cleanliness about the plant. At the present time all make-up water is evaporated, which accounts for the fact that no tubes are now lost in the screens which in this plant are virtually separate boilers. Some trouble was experienced on the first boiler because some of the steel supporting the walls was not properly protected from the heat of the furnace. The design has been changed and this trouble eliminated. The ash from these boilers is removed once a day by sluicing. The ash accumulation in the precipitators and boilers is removed by the same method.

Moisture in the coal used at the Cahokia Station is reduced by drying from 12 per cent to 5 per cent according to the statement of E. H. Tenney, chief engineer of power plants, Union Electric Light & Power Co. The coal used at that station contains about 16 per cent ash, which is high in sulphur and which contains about 18½ per cent of iron. The fusion point of this ash is about 2000 deg. F. The use of this coal has required some experimentation to find a brick which will give satisfactory service. The use of the burners nearest the walls has been discontinued and water cooling of the walls has been resorted to. This plant is near the mines and the expense of economizers was not justified. It handles and prepares its coal for a cost of 36 cents per T. and generates a kilowatt for an expenditure of from 17,223 to 17,950 B.t.u.

Actual operating costs were presented by John Anderson, chief engineer of power plants, Milwaukee Electric Ry. & Light Co. The following costs are for a pulverized fuel burning plant, for the year 1924 and are expressed in cents per ton of coal consumed; operating labor in pulverizing house, up to the boiler room bins, 14.195c; operating labor in boiler room, burning the coal, removing ash, cleaning, maintenance and superintendence, 24.234c; cost of maintenance of fuel preparation equipment, 6.60c; maintenance of burner equipment, 0.94c; maintenance of furnace and setting, 0.793c; electric power consumption, 18.89 kw-hr. per T. and lubrication 0.373c per T. of coal consumed.

During the discussion of these papers, G. C. Bell of the West Penn Power Co. stated that in future designs he would eliminate all offsets in the furnace because they give trouble due to the accumulation of slag. C. W. Clark of Dwight P. Robinson & Co. stated that in changing over the Colfax station to pulverized fuel no serious troubles had been experienced. According to J. W. Parker of the Detroit Edison Co. that company had entered into the use of pulverized fuel not as an enthusiast but rather as a critic and that to date their most serious problem was that of dust, not viewed from the point of its being a hazard, but rather that of cleanliness.

J. C. Hobbs, formerly of the Duquesne Light Co., advocates high furnace temperatures with water-cooled refractories rather than by having the water screens exposed directly to the flame. He stated further that a great advantage of pulverized fuel was the high CO_2 obtainable with low draft while at the same time increased boiler capacity could be obtained thus cutting down the initial cost.

Driers were not installed at the Sherman Creek station of the United Electric Light & Power Co. according to W. A. Caldwell, and they would be used in future designs. This meeting was closed with a statement by J. G. Worker of the American Engineering Co. in which he pointed out that the sale of mechanical stokers followed closely the volume of sales of water-tube boilers.

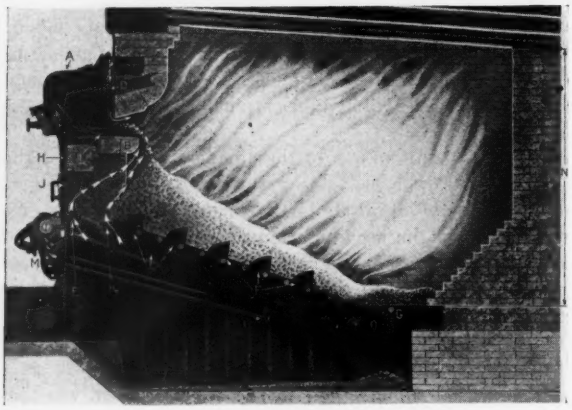
Municipal Power Plant Shows Good Profit

Preliminary figures covering the operation of the municipal electric light and power plant at Anderson, Ind., for the year just closed, show net receipts of $556,644.60, despite the fact that a 10-per cent reduction was made in rates for all classes of consumers during this period. A depreciation fund for the station has been established and maintained, and a considerable amount of the bonded indebtedness has been paid off. The successful attainments of the plant will make for another reduction in rates during 1925, it is stated by Mayor B. J. Horne, as well as the redemption of practically all the bonds now outstanding. Horeover, it is proposed to use a portion of the profits for necessary betterments and alterations in the

plant, with possible installation of additional equipment. Another feature under consideration, is that of the return of interest to consumers on deposits made when service was contracted for.

Coking Shelf a Feature of New Huber Stoker

WORKING on the principle that green coal should first be dried and the light volatile matter driven off, after which the heavier volatiles must be distilled off and the coke broken up frequently, Flynn & Emrich Co., Baltimore, Md., has designed the Huber stoker to be semi-mechanically or mechanically operated.



HUBER STOKER DESIGNED FOR MECHANICAL OPERATION

To accomplish these results, the stoker has been provided with a hopper, which may be fed either by hand or from some mechanical means of coal distribution, such as a weigh larry or from an overhead bunker. Mechanical pushers feed the coal from this hopper A to the coking shelf B, over which is placed a suspended ignition arch C. The passage of the coal over this shelf is slow and continuous over the full width of the furnace. It is here that the coal ignites, due to the radiant heat from the furnace, and is partially coked with the result that the light volatiles are driven off and are burned when mixed with the air which enters through the opening D beneath the ignition arch.

As more coal is fed to the coking shelf, that which has already been partially coked is forced over the edge of the shelf and falls near the front end of the sloping stoker bed. As this ignited, partially-coked fuel drops, it becomes mixed with a further supply of secondary air introduced through the dead plate shutters E and the dead plate bars K.

This fuel which is now coke or fixed carbon is subjected to the full heat of the furnace and by the action of the stoker bars F, which may be either power or manually operated, and whose operation may be automatically controlled for existing requirements, is gradually moved back toward the rear of the furnace. The stoker bar action also keeps the fuel bed continually broken up, shaking out the finer ash and finally depositing the clinkers on the dropbars G at the extreme rear of the furnace where they can easily be dropped into the ashpit. This stoker can be used with either natural or forced draft. The doors are machined and are so constructed that they can be locked tight when it is desired to operate with a blast beneath the stoker.

Details of this stoker are interesting, take for instance, the hydro-motor, which is a simple hydraulic cylinder containing a piston with a range of speed of from 1 to 15 strokes per minute and can be operated from any source of water supply if the pressure is 40 lb. or over. The coal pushers, actuated by the hydro-motor, are designed for adjustable travel which greatly augments the rate of fuel feed which is also controlled by the speed of the hydro-motor.

The elevated position of the coking shelf above the stoker bed permits the building up of a reserve supply of coked fuel, which can be used effectively in meeting peak load conditions. Alternate movement of the stoker bars and the moderate pitch of the stoker bed make it possible to break up the coke pile and to distribute it over the entire surface of the stoker.

These stokers are classified as the "SS," the "SM" and the "FM." The "SS" stoker is strictly a hand-operated stoker and is not provided with a hopper. The type "SM" stoker is semi-mechanically operated, that is, a hydro-motor is provided to operate the pusher bars but the grate bars are hand-operated. Type FM stokers are mechanically operated, being equipped with a hydro-motor to operate the pusher bars and a similar hydraulic motor which operates the grate bars. The cut shown herewith is a section of the mechanically operated stoker.

Lighting Unit for Illuminating Long Aisles

A NEW TYPE of asymmetric lighting unit designed for the illumination of corridors, aisles, passageways and similar places having extreme length in relation to the width, has been designed by the Holophane Glass Co., New York, N. Y.

NEW HOLOPHANE CORRIDOR LIGHTING UNIT

The corridor lighting unit known as No. C-2172, gives an extensive type of distribution along the corridor and an intensive distribution across the corridor. It is designed for 75, 100 and 150-w. lamps to give uniform illumination when spaced on centers not exceeding three times the mounting height. An arrow is molded as an integral part of the reflector which indicates the proper orientation of the unit with respect to the corridor.

This unit is especially suitable for the illumination of aisles between compartments in sub-stations and is an admirable example of how asymmetric light distributions may be moulded to fit specific types of interiors so as to give the greatest possible utilization efficiency of the generated light.

New Speed Transformers Have Herringbone Gears

TWO NEW TYPES of reduction gears have recently been brought out by the Pool Engineering & Machine Co. of Baltimore, Md. These are known as types H and K.

Type H, reduction gear or speed transformer consists of a double helical or herringbone gear made up of an open hearth steel forging and a herringbone pinion cut integral with a high-speed shaft made of chrome vanadium steel.

Both gear members are heat treated to proper hardness to minimize wear. Gear members are accurately

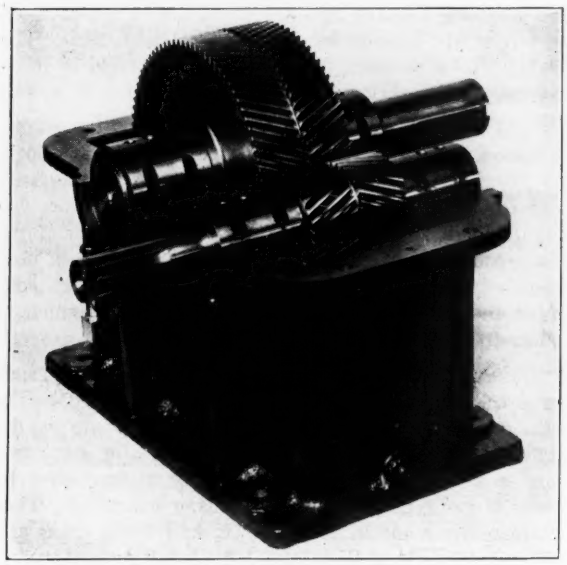


FIG. 1. VIEW OF POOL TYPE H REDUCTION GEAR WITH COVER REMOVED

ground and carefully tested for static balance before being assembled in gear casing.

Gears and pinions are totally enclosed in a cast-iron horizontally split casing with bearing bosses integral with the top and bottom halves of casing. Casing joints are ground perfectly true to insure their being oil tight. The bearings are removable, consisting of cast-iron shells lined with genuine babbited metal and are supported in bosses integral with top and bottom halves of casing insuring accurate and permanent alinement. For low speed drives thorough lubrication is obtained by an improved splash and gravity system in which the gear dips in the oil sufficiently to carry oil to a reservoir in the upper casing from which it feeds to all bearings and directly between the gear teeth at the line of contact.

For turbine and other high-speed drives, an oil pump and cooler is provided, thus insuring cool oil under pressure to the bearings and gear teeth.

The Type K reduction gear consists of double helical or herringbone forged steel pinion integral with shaft and a cast-steel double helical spur gear. Both gear and pinion have cut stub teeth. The gear and pinion shafts are mounted in substantial babbited bearings of the ring oiling type and the gear and pinion are lubricated by the splash system; the gear and pinion running in an oil tight,

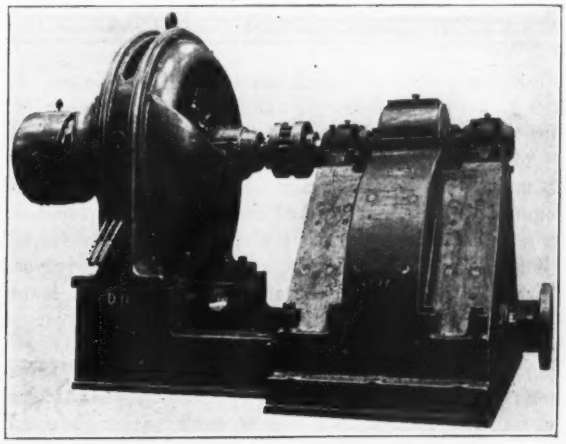


FIG. 2. THE TYPE K REDUCTION GEAR CONNECTED TO MOTOR

oil filled case. The high-speed shaft or driving pinion in the Type K gear is located in the top of the casing and the driven shaft is directly under the pinion shaft and just below or near the floor line which makes it especially desirable for certain types of machines used in industries where the line shaft is close to the floor.

News Notes

FRANCIS C. PRATT, heretofore in charge of engineering for the General Electric Co., is now vice-president in charge of engineering and manufacturing.

WM. H. UTZ, Vice-President and formerly European director of Jenkins Bros., Ltd., with headquarters at London, England, is now director of sales of Jenkins Bros., having assumed these duties on January 1.

ANNOUNCEMENT HAS been made by Freyn, Brassert & Co. that the name of this company was changed on January 2 to Freyn Engineering Co., whose address is 310 S. Michigan Ave., Chicago. H. J. Freyn is president and treasurer, F. H. Wilcox, vice-president and G. P. Pilling, manager of the eastern office, Philadelphia, Pa.

AN OIL ENGINE which will develop 15,000 hp. is under construction by the firm of Blohm & Voss, Hamburg, Germany, for the electric power station of Hamburg. This engine has nine cylinders, is double acting, two stroke and constructed from the latest designs of the Augsburg-Nuremberg machine works. The diameter of each cylinder is 33½ in.

GLENN GRENVILLE HOWE, for many years Senior Vice-President of the Link-Belt Co., died at his home in Muskegon, Mich., on Christmas day after a long illness. Mr. Howe's first connection with the Link-Belt organization was as office boy in 1877 at the age of 16 years. Following his mechanical inclination, he went into the manufacturing department and later became superintendent of the Ewart Mfg. Co. When the Ewart Mfg. Co., the Link-Belt Machinery Co. and the Link-Belt Engineer-

ing Co. were merged in 1906, Mr. Howe became vice-president in charge of the company's Indianapolis operations. He later organized the Howe Chain Co. at Muskegon, Mich., serving as president; but due to failing health, this company was sold to the Link-Belt Co. in January, 1924. Mr. Howe had specialized in chain manufacture and to him are due many improvements which industry is now enjoying.

In the final shipment of electrical equipment, 13 carloads were sent from the General Electric Co. to the Southern California Edison Co. for its 100,000-hp. addition to the Long Beach steam plant. This final shipment was made at about the time the first turbine generator equipment which was shipped in August was placed in service. The total order for the equipment of the station included two 35,000-kw. turbine generators, each equipped with a 2800-kw. house generator, ten 13,000-kv.a. transformers, eight 73,000-v. lightning arresters, twenty-eight 73,000-v. 40-amp. oil circuit breakers, twelve 73,000-v. potential transformers, sixteen 73,000-v. current transformers, two complete sets of air coolers, one 350-kw. motor driven exciter, two 175-kw. motor driven exciters, three 250-hp., 2200-v. motors, one 175-hp. 2200-v. motor, five 150-hp. 2200-v. motors, eight 100-hp. 2200-v. motors, two 75-hp. 2200-v. motors, and approximately thirty 5 to 40-hp., 440-v. motors, control equipment for handling all of the above motors, electrically operated steam flow meter equipment, oil filters and dryers, battery charging motor generating sets and other miscellaneous equipment.

Catalog Notes

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"Applications of Leather Belting" is the subject of a 24-page book prepared by the Leather Belting Exchange Foundation, giving a non-technical summary of research work. The book covers a large number of experiments conducted in the laboratory of Cornell University on pulley diameters, pulley ratios, center distances, the effect of high belt speeds, the gravity idler, from which R. F. Jones, the research director, has developed new rating curves for leather belting providing the necessary correction factors with their values. The book may be obtained from the Leather Belting Exchange, Philadelphia, Pa.

Sparks from the Advertising Pages

Where Buyers and Sellers Meet

The Fittest Survive

Concede the theory that all men are born free and equal and we at once find ourselves up against the incontrovertible fact that few of us remain either free or equal for any appreciable length of time after birth.

Engineers realize that it is difficult to make two boilers, stokers, engines, turbines or two machines of any kind that are exactly alike. Turned out of the same shop at the same time, built side by each, there rarely are two mechanical units that do not develop unexpected differences from the time they are assembled and tested to the time they are junked.

To expect to successfully make human beings free and equal presupposes that they must be at least created as nearly in the same mold as twin pieces of machinery.

The United States enjoys the distinction of working out freedom and equality to a degree beyond that attained in any other country.

In men it has so far been largely political freedom and equality; in machines freedom from trouble and equality of performance has been developed through standardization.

It seems self-evident that with men as with machines, standardization must always precede freedom and equality.

In truth this is the very process by which we have gained our political freedom.

Our Declaration of Independence, our Constitution and all the laws, are each a standard, each a measure or mode of conduct for individuals or institutions.

And they do not bestow freedom except as they maintain restriction, for no law or standard is enforced or maintained except to the extent that individuals conform to the mold set by the law.

To find out how to increase the welfare of all the people let us consider the way in which a machine is made.

This is a slow process but a sure one. An idea is conceived for an invention. Drawings are worked out and a model constructed. Financing, manufacture, testing, installation, selling and continual building in time establish success.

Throughout the course of this road to success scores, hundreds and even thousands of men may be drawn into more or less intimate contact with this machine.

Each contributes ideas, money or workmanship until the machine is a composite of the thought and effort of a group of men.

Then, as this process goes on, there must arise the groups of men who buy, operate and profit by this machine.

Beyond these are greater groups, ultimately the entire population perhaps, the people who buy the products of this machine, or whose economic status will be affected in some measure by the success or failure of the primary groups who make it and use it.

In this process is evolved the machinery, equipment and supplies offered in these pages. Through a literal survival of the fittest—the fiercest of conflicts—they have reached the point where they are offered for your service.

Only such a process could turn out safe, efficient, reliable products on which you in turn may depend for results.

How could it bring any improvement to the power plant industry to disturb the fine balance of this process by which improvements are worked out?

You would not want to buy boilers that were built in shops where all men were free and equal in the sense that only too many men would like to be.

Guaranteed a living, with no incentive or compulsion either to improve or maintain standards, and with no consequent feeling of pride or responsibility, we can all imagine what the product would be like.

. We can as readily imagine a whole world run the way of this boiler shop.

No man trained to deal with facts as they are, and to take responsibility, needs any explanation of what a world this would be without incentive and the will to fight things out on a competitive basis.

Nor does any real doer in the world care for a new order of things or a new deal. He is willing to play the cards as they lie and improve the rules a little at a time as fast as the game grows stronger and the majority of the players come to live up to those we have been playing under.

To abolish the fundamentals simply means that there would be no possibility of anyone winning; soon there would be no game; in time, no players.

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